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SMALL ARMS SYSTEMS LABORATORY

TECHNICAL NOTES

SMALL ARMS WEAPONS DESIGN

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SMALL ARMS WEAPONS DESIGN

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I Introduction

The topics discussed in this manuscript were included in a series of lectures conducted by Mr. John G. Rocha, formerly of Weapons Development Branch, Research and Engineering Division, of the Springfield Armory. The lecture series was given at Rock Island Arsenal, Rock Island, Illinois during the period October 1967 - March 1968. Preparation of these lectures was directed toward assisting ordnance engineers of the newly formed Small Arms Research Branch of Rock Island Arsenal in becoming knowledgeable as to ordnance design practices commonly known at Springfield Armory.

A number of selected ordnance design publications and tests were used as source data, and is noted in the bibliography, as well as other recommended reading. In addition to the published source material, available in most ordnance engineering libraries, observations made during Mr. Rocha's 15 years experience as ordnance design engineer at Springfield Armory are included throughout the manuscript.

Included in the lecture series, following a historical resume, are topics on interior and exterior ballistics, systems of weapon operation, stresses, dynamics, kinematics of mechanisms, and topics peculiar to ordnance engineering, such as headspace, recoil forces, links, magazines, and evaluation of time-displacement curves.

Emphasis is placed upon the coordination required between weapon design and ammunition design agencies since the weapon system demands that each group be cognizant of weapon/ammunition interface areas. In addition, this writer feels that any new weapon design, or weapon system reflecting a quantum increase in firepower, must depend upon increased effectiveness of ammunition. Rare exceptions are the externally powered high rate of fire machine guns that increase volume of fire per installation. Advances in future weapon effectiveness may well be triggered by improvements or new concepts in ammunition design. Therefore, again, the weapon designer should be knowledgeable of all the facets of ammunition design including chemistry, ballistics, and thermodynamics, as well as cartridge case and projectile design. "Ballistics" includes interior, exterior, and terminal, or wound, ballistics. In an ordnance organization, only one or two engineers will be thoroughly familiar with any phases of these allied sciences, and will function as a consultant to the section on the particular topic in question.

The material discussed in these lectures should be of interest, and informative to all levels of a small arms design section; but further specialized data on each topic will be left to the initiative of the user, since ordnance engineering is a viable science, and new discoveries are constantly outdating the status quo.

II Historical

Ever since the caveman used a club or rock as the first weapons, man has continued to improve the design of his weapons, often with the hope that the new weapon developed would end all wars and bring peace for all mankind.

As an example, in 1704, a French engineer named Chaumette devised a new approach to solving the problem of loading a rifle from the breech quickly and effectively. A presentation model of this weapon had the following inscription engraved on the Barrel:

> "La Chaumette has made this terrible gun. All its patrons will be blessed for it is the means of ending war and establishing the Golden Age."

Doctor Richard Gatling interrupted his medical practice during the 1860's to develop his famous multiple barrel weapon, which principles are still prominent today. He confided to his associates that his weapon would put an end to all wars and bring peace to the world. Unfortunately, he was mistaken, because the development of any new weapon spurs the development of a counterpart.

Historically, Past Is Prologue, and this is significant in many weapon designs. A prodigious number of weapon mechanisms have been introduced in the past that failed, but would be adaptable to future weapon designs if the principles are correctly applied. This is because of the many advances made in metallurgy, new alloys, new production techniques, and propellants. For example, the hexagonal bored rifle (Whitworth) of over 100 years ago is being re-introduced in the U.S. in 1968 (commercially) as rifling "without lands or grooves".

Development of Military Cartridge Case

Friar Bacon was one of the first Europeans to develop a successful formula for gunpowder. At this time, practitioners of the art were more concerned with the noise and flash produced, and observed that confinement of the charge accelerated pressure build-up, but it did not occur to them that this pressure could be utilized to propel shot. This activity started about the year 1250. Another monk, Berthold Schwartz, used gun powder in the period of 1290 - 1350. He experimented with heating sulphur, saltpeter, mercury, and charcoal, attempting to convert mercury to silver, and succeeded in flattening himself and the laboratory in several explosions before he understood the principles of propulsion. The Chinese also used black powder, in the 6th to 10th centuries, but only for ceremonial and demonstrative purposes.

Mechanical art in various forms dictated the course of warfare thereafter. The first weapons were cannon (from the Latin "CANNA") because of their reed-like construction, and the first projectiles were, not balls, but arrows.

For the next few centries, ignition systems were the principal design keys. Initially, the gunner had to ignite the powder through a blow-hole on top of the barrel, so that he could not aim at the same time. Accuracy, then, was only accidental. The touch-hole was moved from the top to the right side of the barrel. A little ledge, or pan, was added beneath it to hold the priming powder and thus make ignition more certain. A hinged cover was added to protect it from the weather. Barrels were lengthened, stocks were shortened and the general contours of a modern gun began to appear. Most important was the development of a wick, or match, together with a device for holding it. This was a twisted rope dipped in saltpeter and spirits of wine, so that it burned slowly, as a punk; thus the shooter did not have to stay near the campfire to ignite his weapon. Thus mobility was improved.

The complete gun-lock, stock, and barrel, appeared with the match lock, which was a spring-loaded device that brought the match into the priming mix when the trigger was pulled.

Bullets developed slowly. Hand cannon fired not only lead, but also stone, iron, steel, brass, copper, and tin missiles as well. As this was an age of experimentation in ballistics, there were also cylindrical, pyramidal, rectangular, and barrel shaped bullets. Arrows were also popular, even silver buttons on one historic occasion. By 1600, the lead ball was universally used. The match lock, despite its limitations remained in usage until the 1700's. About that time, wheel-locks were devised, which worked as simply as the cigarette lighter works; that is, friction between a serrated wheel and pyrite produced sparks to ignite powder in the pan.

The first known reference to a cartridge was by Leonardo da Vinci about 1500, a simple tube of rolled paper, each holding powder for one shot. The gunner simply bit off one end, poured a little powder in the pan and the rest down the barrel. The ball followed, and the paper as a wad. Rate of fire thus improved.

Then came a whole group of ignition systems that produced sparks by striking flint against steel. This improved maintenance, as the wheel-lock mechanisms were complex. The flint-lock, or snap haunce, mechanism was simple. The classic flint-lock mechanism was designed in France in the early 1600's and carried on for at least two centuries, the most noted being the British Brown Bess and the French Charleville. At this time firing rate was about 4 SPM.

The flintlock mechanism also made the pocket pistol really practical.

One big improvement by Henry Nock, was the plenum (or pre-ignition) chamber, to speed ignition, improve maintenance, and improve ballistics.

A clergyman, Alexander Forsyth ushered in the modern era with the first successful percussion lock. This was the turning point in the history of firearms, as it provided the basic theory for all future developments in ignition, including the modern metallic cartridge.

Forsyth's hobby of hunting led him to his remarkable discoveries. He noticed that many of the wild birds escaped his fire by diving the instant they saw sparks or flash from his lock. The ensuing hesitation in ignition was all they needed. He set about to re-develop the firearm in the late 1700's to remedy this inefficient ignition system. New substances being experimented with at that time were called fulminates, or salts produced by dissolving metals in acids. When struck, they exploded violently. After much experimenting, Forsyth succeeded in 1805 in directing the priming charge into the bore. With the help of James Watt, he patented his discoveries. Many variations of his principles led to the percussion cap. This cap was placed on hollow steel nipple, and, when struck, directed a flash into the bore.

The origin of rifling dates back to the early 1500's. Longitudinal grooves were cut in the bore to collect powder residue. When someone spiralled the grooves to increase their length, accuracy improved to everyone's surprise. When tight-fitting projectiles were used, this led to problems in loading, until greased patches were introduced. This also helped clean the bore.

However, with improved accuracy, speed of loading was sacrificed, by the time taken to drive the bullet down the bore in loading. A French army captain, Claude Minie, refined the shape of some experimental hollow-based bullets in 1849 so that, via a iron cup, the projectile skirt expanded into the rifling. Hence the "Minie Ball" was born and proved to be highly effective during the (so-called) Civil War.

Soon afterward, in 1855, an American dentist, Edward Maynard, developed a tape primer, similar to the common children's "caps" in roll form for toy pistols. Hence, a method for speeding the reloading of the ignition system was devised. It soon became clear that it was easier to load a gun at the breech rather than putting a charge of powder, ball, and wadding down the bore, particularly when lying down, confined or under combat stress.

Breech loading had been experimented with for several centuries dating back to the early 1500's when shield-type pistols, of necessity, were breech loaders, using an iron tube similar to the modern cartridge case. Firing was by a match-lock.

The first important breech-loading military weapon was French, and utilized a threaded breech plug perpendicular to the bore, which did not fall free, and was thus easy to load. (1720's)

A large number of gun enthusiasts proceeded to develop a prodigious number of breech loading variations, but powder fouling, gas leakage, and endurance limited any success for well over a century. Finally, Pauly, a Swiss inventor, initiated the concept of using the ammunition as the key to breech-loading design, in 1812, when he used a cartridge with a rimmed head of soft metal, to obturate the propellant gases. These cases had paper bodies with brass heads, like common shotgun shells. One of his technicians, Johann von Dreyse, became one of the most historic figures in the development of breechloaders, as the inventor of the bolt action rifle with the "needle gun" principle of ignition. The basic principle was that of a priming compound in a hollow at the base of the bullet. The primer was detonated by a long firing pin that pierced the powder charge.

With this gun, the Prussians dispatched the Danes quickly, and the Austrians in only 7 weeks, in 1866. Additional developments were made to shorten and strengthen the firing pin, which proved to be a problem in erosion and endurance.

Christian Sharps made another important contribution with his lever action carbine. He utilized a separate-disc priming mechanism that automatically positioned a primer over the nipple for each shot. This weapon was prominent in the Civil War, as well as in pioneering and settling the West. It was the last of the important combustible case weapons. Cartridges were made of paper, linen, rubber, metal foil, and sheet metal, in a variety of breech mechanisms. The basic problem of all breech-loaders was the same: that of obtaining a quick-acting gastight seal in a mechanism that could function for a follow-on shot.

The pin-fire type of cartridge was briefly successful, but proved to be fragile, and prone to accidental firing. From the side-firing pin-fire followed the teat fire and the rim-fire, which was limited to low-powered cartridges, because the metal had to be hard enough to support the chamber pressure, yet soft enough to be easily indented by the hammer or firing pin. The center-fire cartridge eliminated this limitation. In this, a primer pellet is crushed by the striker against a rigid anvil, with vents leading into the main charge. Two important types, by Col. Berdan, U.S., and by Col. Boxer, U.K. were developed and have been essentially unchanged in 100 years.

The 1873 Springfield .45-70 trap-door model was simply a means of converting stock-piles of muzzle loaders into breech loaders. One problem proved to be in tight extraction after prolonged firing due to thermal expansion of the breech end of the barrel.

During this period, in the U.S. and U.K., 120 actions and 50 cartridges were considered in a search for most acceptable mechanism, attesting to the variety of inventions produced in the 1870's, following introduction of the modern metallic case.

In 1889 Alfred Nobel invented "smokeless" powder, Vielle of France developed the ballistics theories, and the age of high performance weapon systems was born.

The United States Light Rifle Program

This section is a resume of the activity, generally at Springfield Armory, directed at satisfying the User's requirement for a lightweight automatic shoulder weapon. This requirement was a natural follow-on to the semi-automatic (8 round) M1 rifle which gave the U.S. a significant advantage over other countries that were armed with bolt action rifles.

Modifications to the M1 system were initiated in 1944 as the Springfield Armory T20 series and the Remington Arms T22 series. These developments included use of 20 - round box-type magazines selective five with semiautomatic fire on "closed bolt" and full automatic fire on "open bolt". ("Open bolt" means that the bolt is held in the full recoil position between firing bursts.) (This is done to minimize the danger of a cook-off, the inadvertent firing of a cartridge caused by the cartridge being in contact with a hot chamber. Ingition is by conduction of heat through the brass case and into the propellant. The primer is not struck!) Eventually the requirement for the "open bolt" sear was deleted.

Experiments also included use of cut-off and expansion gas systems, muzzle brakes, compensators, bipods, and a number of minor items.

The initiation of the formal light rifle program was featured by a shorter cartridge, the T65 series, which eventually was developed into the 7.62mm NATO standard cartridge. The Cal. 30-06 cartridge (used in the bolt-action Springfield '03 as well as the BAR and the M1 rifle) is 3.33 inches long, while the NATO cartridge is 2.80 inches long.

One of the first rifles to take advantage of the shorter cartridge was the T25 rifle, a Springfield Armory design. It featured a tip-up bolt lock not unlike the BAR lock, an expansion gas system, and an inline butt-stock to facilitate control in automatic fire. User preference in styling dictated redevelopment of this model as the T47, with a conventional drop-stock.

Another Springfield Armory development was the T28 light rifle, which featured a modification of a German conceived breech mechanism, extensive use of stampings, and an inline buttstock, also to assist in control of automatic fire. The locking rollers lock the bolt to the barrel extension, as typified in the German MG42, the Spanish CETME, and the current German G3 rifle. However, these European models are engineered to utilize a retarded blowback principle of operation to cycle the weapon. This does not allow a reserve of

power in the event of firing under adverse conditions, so the T28 rifle was developed with positively locked rollers and a gas system of operation to power the bolt carrier.

The T44 series of weapons was basically a continuation of the T20 series of M1 rifle modifications. Eventually, this model (T44E4) was standardized as the M14 rifle.

The T31 rifle was an experimental model designed by John Garand which was characterized by an inline stock, "bull pup" configuration, a pistol grip in front of the 20- round box magazine, and a number of unconventional promising features. These included, among others, a simple leaf-type driving spring, a firing mechanism that utilized only a short stroke of the operating rod, and a gas-operated barrel cooling system. If this weapon had been continued in development, it could well have minimized the necessity for use of the M16E1 rifle. Of course, this is a speculative statement, but it demonstrates how "timing" in the introduction of a novel weapon is important. The User must be prepared to accept a seemingly unconventional item. User preference in styling "feel" (balance), sighting geometries, and other intuitive qualities cannot be technically measured.

The T35, T36, and T37 models were interim phases between the T20 and T44 series.

The T48 rifle was the Belgian FN entry in the competition for selection of a NATO rifle. It was also chambered for the T65E3 cartridge, as was all of the models in the preceding paragraphs. A pilot line of 500 models was manufactured in the U.S. by H.&R., in order to measure adaptability of the design to U.S. production and inspection techniques. Selection of the standard U.S. rifle was eventually based upon a series of U.S. User tests.

A series of feasibility studies for an advanced weapon system evolved as the "Salvo" concept. This was based upon the observations made under combat conditions of the shooters level of ability, or inability, to hit his target. There are many facets to this problem, but it can best be summarized as effort to improve "hit probability". Weapon developments centered about test fixtures using multiple barrels, for simultaneous, or ripple-fire multiple shots per trigger pull. The SPIW rifle program is one generation of this concept study. The 7.62mm Duplex cartridge is one result of this program.

The SPIW system is a shoulder weapon that fires a lightweight flechette projectile saboted in a smooth-bore 5.56mm barrel. The firing mechanism is designed to permit single-shot semi-automatic, high-rate three-round-burst semi-automatic, or full automatic fire. The lecture on "Dynamics or Automatic Rifles" will demonstrate that the high-rate three-round-burst is optimum commensurate with controllability and hit probability.

Studies of liquid propellant rifle systems are discussed in an ensueing chapter.

An additional rifle development was the Springfield Infantry Rifle, designated as "S.I.R." This was a lightweight, compact, Cal. .224 automatic rifle that was designed to incorporate the best features of all of the infantry rifles developed to date, and using a lightweight cartridge that preceded the present 5.56mm round used in the M16E1. This program was also derivative of the "Salvo" concept.

Designers, or principal engineers, of the weapons disclosed in this chapter are as follows:

T20 series	John C. Garand
T25	E.M. Harvey
T28	C. A. Moore
T44	L. S. Corbett
T31	John C. Garand
SPIW	R. H. Colby
S.I.R.	A.J. Lizza

An excellent description of U.S. standard machine guns is given in Smith & Smith's eighth edition of "Small Arms of the World" as well as in G.M. Chinn's Vols I - IV of "The Machine Gun".

<u>Caseless Ammunition</u>

Now after the 100 year period since cartridge cases were introduced, efforts are being made to eliminate them, taking advantage of the growth of other technologies, such as metallurgy and chemistry.

In chemistry, new forms of caseless ammunition have been developed, such as a solid molded propellant containing the projectile and a combustible percussion primer. In metallurgy, new and improved materials may lead to successful methods of obturation, the reduction of heat transfer, and elimination of erosion and cook-off.

What are the advantages of this new facet of ordnance design? Essentially, they are as follows:

1. COST - The price of a cartridge case is approximately one half of the cost of a complete round. Eliminating the case should cut ammunition cost in half. (However, note development effort required to realize this)

2. WEIGHT - The weight of a brass case is approximately one half of the weight of the complete round.

3. BULK - The molded caseless cartridge is approximately 25% shorter than the cased ammunition.

4. ELIMINATION of spent cases that clutter the compartment of a vehicle, particularly larger caliber weapons.

5. Possible simplification of gun mechanism, such as eliminating close tolerance headspace dimensions.

6. Critical cartridge brass material will not be required.

The weight and bulk factor are especially attractive from a logistics and installation point of view.

For example, in comparing the 7.62mm NATO cartridge with a comparable 7.62mm caseless round (of comparable ballistics), the Nato round weighs 390 grains and has a storage volume of .85 cubic inches, while the caseless round weighs 196 grains and has a storage volume of .64 cubic inches. The effect of this is that an infantryman's ammunition load (220 rounds) is reduced from 15 lb. to 8.7 lb., or, if the same combat load is used, the caseless ammunition complement is increased to 430 rounds.

For armament installations, the advantages are particularly attractive. In helicopters, as in all aircraft and most vehicles, weight and space are at a premium. A typical installation, the M6 Armament Subsystem, (Quad 7.62mm M60C machine guns) carries 6000 rounds, with a weighting of 390 lb. and a volume of 3 cubic feet. Caseless ammunition would weigh 190 lb. and have a volume of 1.95 cubic feet. If the same system weight is held, the ammunition complement could increase from 6000 rounds to 11,500 rounds.

So much for the prospects, but now what are the problems? This is where the present and future development effort is concentrated, and is generally as follows:

1. COST - Establish production techniques so that ultimate mass production at low cost will be realized.

2. CONFIGURATION - The proper size and shape of caseless ammunition commensurate with the following:

   a. Production techniques and control

   b. Weapon design (close coord. req'd)

   c. Ballistic efficiency

   d. Feeding, storage, and handling

3. OBTURATION - Chamber, bolt, firing pin

4. COOK-OFF (Cartridge case had functioned as insulator)

5. EROSION and fouling

6. MISFIRE extraction

7. COATINGS against moisture

8. STRENGTH to support handling & feeding (durability)

Of the above problem areas, obturation, or the sealing of the high pressure gases from excessive leakage, is expected to be the most articulate if endurance is expected. That is, a self-acting durable seal for an automatic weapon will not be merely a close-tolerance fitted component, because prolonged firing will cause thermal expansion of components to varying levels.

Some relief from the wide scope of problem areas inherent in a caseless system may be realized by using a partially combustible cartridge case. This is done in the 105 mm howitzer and 105 mm gun area. In this, a stub brass shell is used, in order to provide obturation, ease of extraction, etc. etc. A significant reduction in cartridge brass is realized, being approximately 85%, rather than the 100% of completely caseless ammunition.

Liquid Propellant Systems

Approximately 10 - 12 years ago a considerable amount of work was done in exploring the feasibility of utilizing liquid propellants in small arms. The range of programs was quite extensive, and enveloped a number of applications and philosophies. Initially, it was held to be promising for ship-board installation, where the propellant bulk could be stored in remote areas, and pumped to the weapons. This concept then was applied to tank guns, then interest grew in the small arms field.

There are two general categories of liquid propellant systems: the monopropellants and the bi-propellants. In the bi-propellants two separate liquids, a fuel and an oxidizer, are pumped into a chamber where hypergolic action causes ignition and burning. No primer or igniter is used. Monopropellants are single fuel systems, with ignition by separate means, such as spark, primer, compression, glow-plug, etc.

The advantages of a liquid propellant system are generally as follows:

1. Greater impetus per pound of propellant for liquid (over solid propellant) systems.

2. Sharp reduction in pressure peak through control of pressure-time interior ballistics.

3. Hyper-velocities possible (over 5000 fps projectile).

4. Reduced heat transfer to barrel and chamber.

5. Cleaner burning (no fouling).

6. Reduced erosion.

7. Elimination of cartridge brass.

8. High capacity weapon systems may be realized.

9. Reduced smoke and flash.

10. Reduced recoil peak loads (inherent with item 2 above).

11. Remote storage of propellant (for vehicular installations) to save space in gunner's compartment, and for safety.

12.. High firing rates possible in certain weapon designs.

During the course of development, it was realized that the high impetus propellants were extremely corrosive and unstable. Erratic pressure peaks and erratic ignition & burning ensued, until eventually the bulk of effort was directed toward monopropellant systems in which the propellant was a mixture of hydrazine, hydrazine nitrate, and water. (approx 70-25-5) A typical bi-propellant combination would be Red Fuming Nitric Acid and Hydrogen Peroxide, for example, there being a wide range of propellants considered.

Two philosophies in the liquid monopropellant field centered about the control of the pumping, or chamber-filling, process. One was a constant-pressure theory, the other a constant-volume theory.

In the constant-pressure system, the chamber was filled until a definite liquid pressure of the full chamber was attained. Here, as firing progressed, any change in bullet seat would change bullet position, increasing the propellant volume, causing a change in interior ballistics.

In the constant-volume system, a definite, measured amount of liquid was pumped into the chamber. As firing progressed, any change in bullet seat would cause ullage, or an amount of air, in the chamber, which caused erratic ignition and uncontrolled interior ballistics performance.

In general, the following problem areas remain unsolved and constituted the principle R & D effort:

1. Obturation, both at low (fill) pressure and high (burning) pressure

2. Ignition

   a. Erratic

   b. Energy source for spark

3. Purging system after misfire

4. Pumping

   a. Cavitation

   b. Pre-ignition

   c. Leakage

5. Erratic chamber pressures

6. Poor low temperature characteristics of propellants.

III Supporting Sciences

This section reviews some of the fundamentals of interior and exterior ballistics, recoil forces, and dynamics of automatic rifles from a mathematical, rather than hardware, perspective.

It is felt that design engineers should be knowledgeable of the sciences that affect their product.

Much of the data indicated here may be corroborated by test fixture firings, since the results are limited to specific calibers, weights, velocities, and other characteristics. These fields are constantly in need of revision because the formulae involved are limited to the specific conditions of the tests observed. This is one of the reasons that the science of ordnance engineering is so fascinating.

Interior Ballistics

An understanding of the fundamentals of interior ballistics is essential for the weapon design engineer. This is because the

development of pressure in the chamber and bore affects the weapon, operating system, and is a basic weapon/ammunition interface area. The weapon designer should be prepared to make positive recommendations as to ballistics parameters when a new weapon system is being specified, or is in a state of development.

The following are examples of weapon/ammunition interface areas that requires knowledge of interior ballistics by the weapon designer.

a. Headspace limits,

b. Timing breech opening,

c. Development of stresses on barrel bolt lugs and breech ring,

d. Rifling twist,

e. Barrel and chamber erosion,

f. Gas pressure at orifice (for gas operated weapons),

g. Gas pressure at muzzle (for recoil operated weapons) for design of muzzle booster,

h. Construction of primer to prevent primer puncture and/or cup flow,

i. Construction of cartridge case to prevent case splitting,

j. Reduction of smoke and flash, as well as various products, and

k. Ballistic stability at temperature extremes.

A diagram of a typical Cal. .30 pressure travel curve is analysed and illustrates pressure and velocity with respect to bullet travel.

This typical cartridge fires a 150 grain projectile at a muzzle velocity of 2700 feet/second. Quite simply, the formula for projectile muzzle energy is: $E = WV^2/2g$.

$$E = \frac{150 \times 2700^2}{7000 \times 64.4} = 2420 \text{ ft. lb.}$$

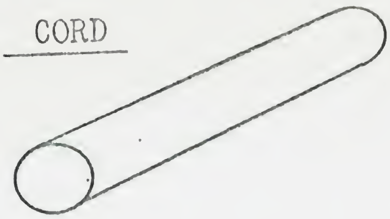
Correlate this with integration of the pressure-travel curve: That is, the area under the curve multiplied by the bore area is equal to the energy developed.

Travel scale 1/4" = 1" (typical)

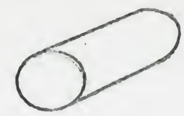
Ref.: Hatcher's Notebook by J.S. Hatcher, Chapter on Interior Ballisti

COMMON POWDER TYPES for SMALL ARMS

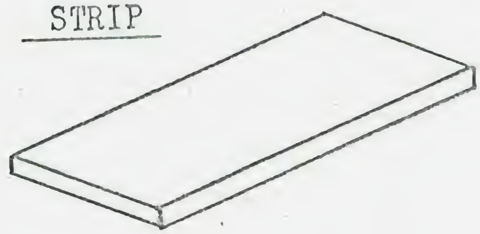
CORD



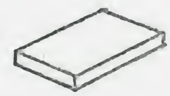
PELLET



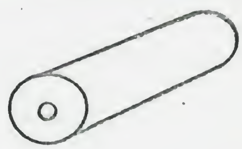
STRIP



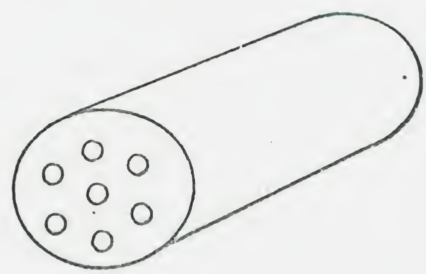
FLAKE



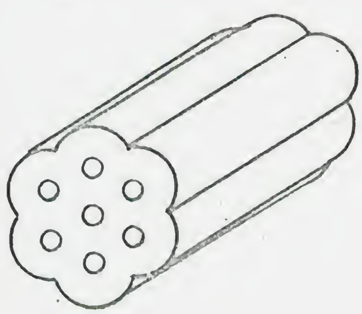
SINGLE PERFORATION



MULTIPLE PERFORATION



ROSETTE



BALL

FLATTENED BALL

NOTE: The pressure the propellant develops varies inversely with the thickness of propellant grain.

Pressure scale 1/4" = 5000 psi.

each square = $1/16"^2$ = 5000 in. lb./in.^{2}

Count of squares = 78 (use of grid)

78 x 5000 = 390,000 in. lb./in.^{2}

Bore area = $\pi/4$ $.308^2$ = .0745 in.^{2}

Energy = 390,000 x .0745 = 29000 in. lb.

E = 2,420 ft. lb. Therefore, the data correlates.

The identical procedure is utilized with a pressure-time curve, in which the unit is lb.-sec., or a measure of impulse.

Note that if a heavier projectile were used, which would result in increased muzzle energy, or a more powerful charge were used to increase velocity, then the area under the curve would have to increase proportionately. Since barrel travel is constant, then the pressure would have to increase. This is one penalty that is paid for packing more propellant into the cartridge case.

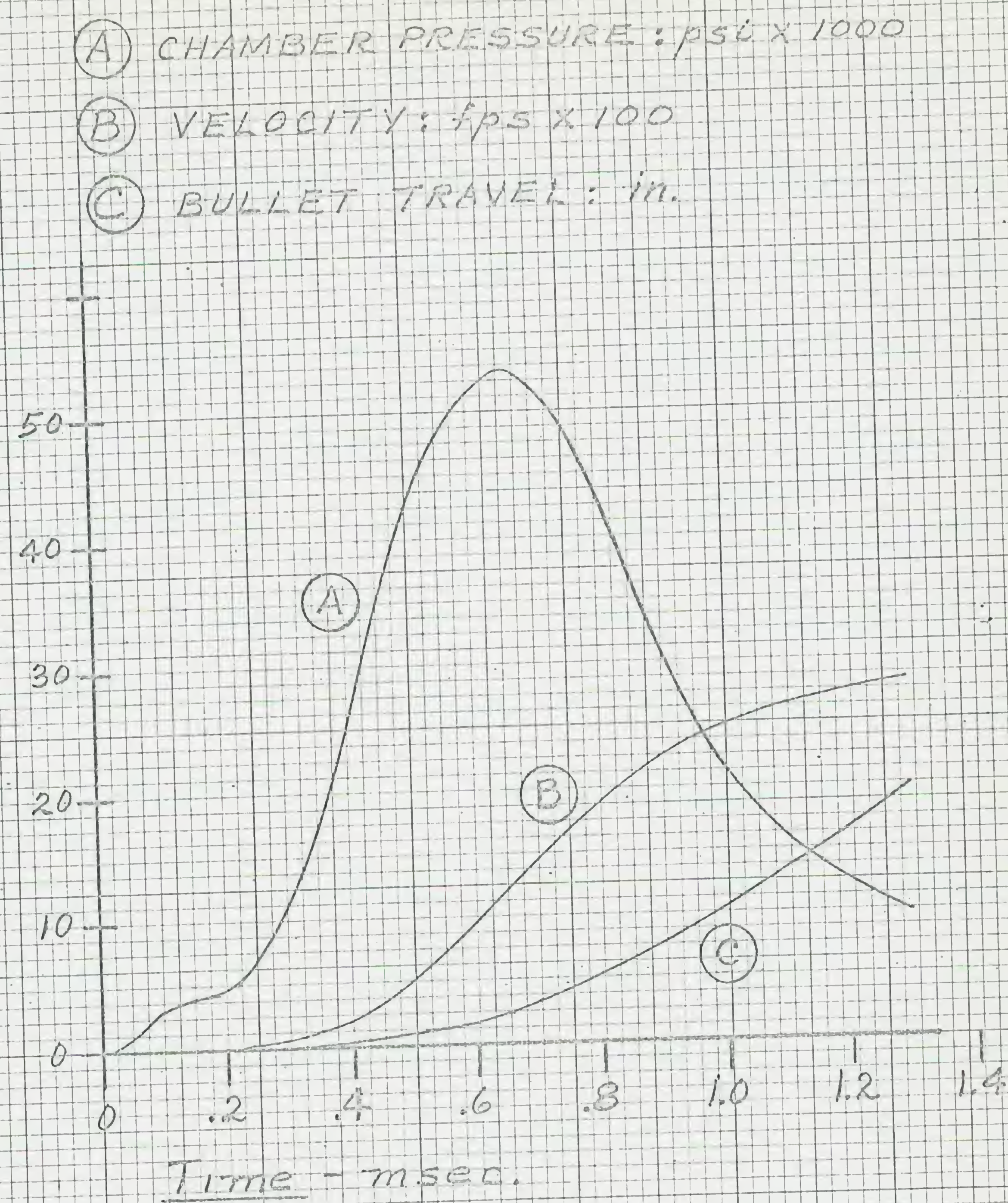
At times attempts have been made to alter the shape of the pressure curve, so that a lower peak, acting for a longer time, would result in reduced weight of locking mechanism barrel, etc. but there are limits to the effect of such a change.

The development of pressure in the weapon is also controlled by simply changing the chemical composition of the powder or by changing its form. The pressure the propellant develops varies inversely as the thickness of the propellant web. Smokeless powders are most commonly used, being a double-based composition of nitrocellulose (gun cotton) and nitroglycerine. When completely decomposed, nitrocellulose yields CO_2, CO, H_2O, N_2, and nitroglycerine yields CO_2, H_2O, N_2, and O_2. The exploding temperature of both materials is about 180°C. -200°C.

Note that the propellant contains its own oxygen.

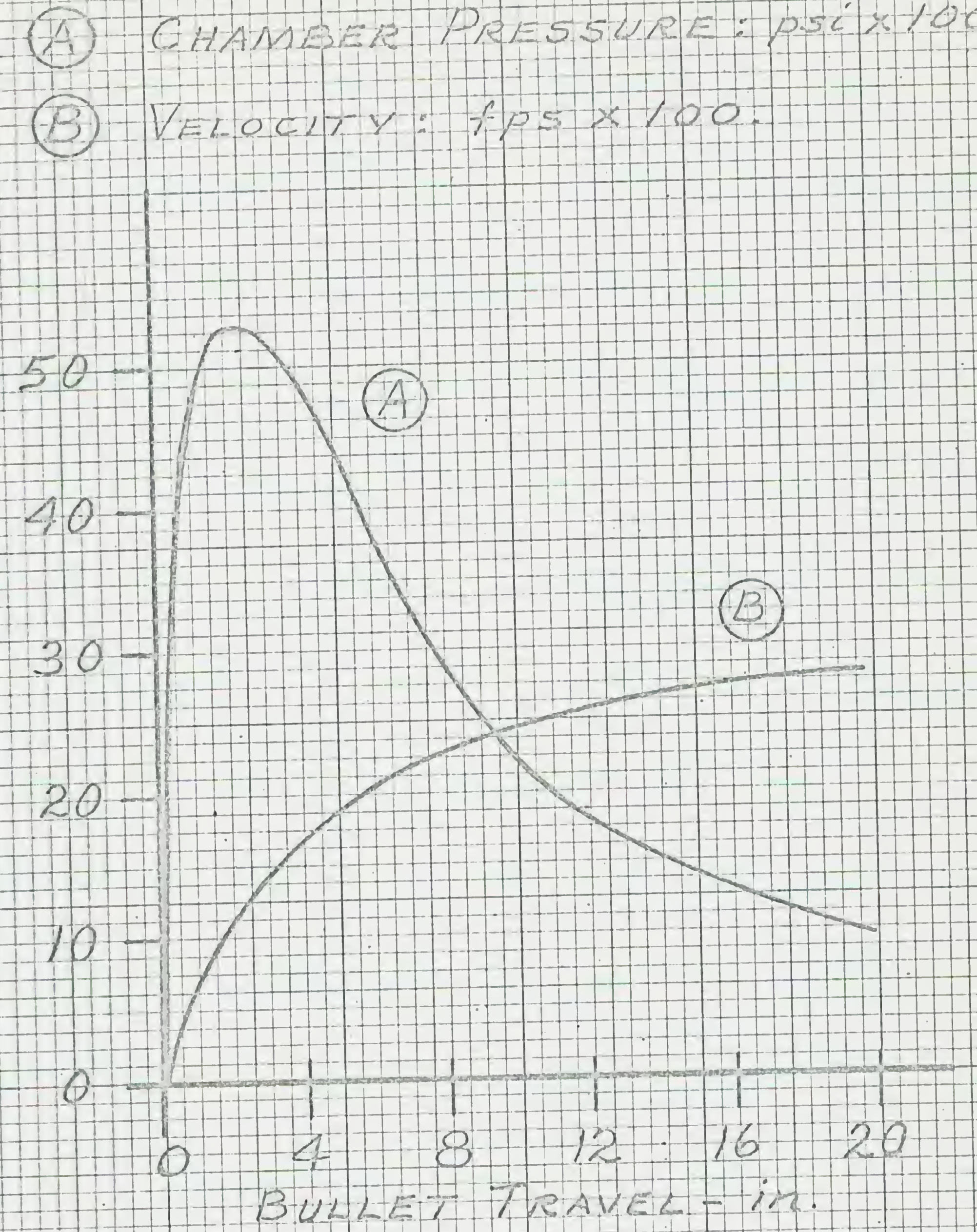
A material advantage of smokeless powder over the old black powder is the fact that it burns in parallel layers. Therefore, by appropriate shaping, the burning rate, and thus the rate of pressure increase, may be controlled.

Moisture must also be controlled in nitrocellulose powders. A change in moisture content of ± 1% changes the muzzle velocity by ∓ 12 fps and the gas pressure by ∓ 750 psi.



TYPICAL INTERIOR BALLISTIC DATA

7.62 mm NATO (BALL)



INTERIOR BALLISTICS DATA

PRESSURE & VELOCITY VS. TRAVEL

7.62 mm NATO (BALL)

Temperature change also changes gas pressure and velocity accordingly. The propellant is ignited by the primer charge, either fulminate of mercury or lead azide, so that a large flame is created by adding a very surface-rich powder, so that practically all the powder grains begin to burn at the same time. The rate of burning is a function of the pressure of confinement. The speed of reaction increases with increasing pressure until a pressure and temperature maximum is reached, producing an explosion. Loading density, therefore, is an important value in internal ballistics. Loading density is the ratio of propellant charge to chamber cavity volume, in grams per C.C.

Improper selection of propellant chemistry could result in a high temperature increase at concentrated points in the propellant mass, causing a pressure wave to emanate, that travels through the powder mass in a shock wave, creating a detonation.

In order to guarantee consistent burning from round to round, and thus consistent pressure development, the chamber density should not exceed a maximum value, which depends upon the burning heat of the powder used. The loading density, therefore affects uniformity of weapon performance.

Development of formulae on Internal Ballistics has been principally through the efforts of Vielle (France), Granz (Germany), and Charbonnier (France).

The burning temperature of nitrocellulose powder will peak, in a typical weapon, at 3000°F, with the temperature at the muzzle exit of 2000°K.

It has been proved mathematically that the maximum attainable muzzle velocity of a projectile, using nitrocellulose powder is 9100 fps. This considers 36% of the charge transformed into muzzle energy of the projectile and charge. This figure is only of academic interest, because it requires extremely high charge to mass ratios. "Charge to mass ratio" is merely the propellant weight/projectile weight, and for small arms varies from .3 to .4. Higher propellant loads are inefficient, causing temperature and barrel erosion values to accelerate.

A typical variation of muzzle velocity with powder temperature change is as follows:

Temp. degrees F.	Velocity
0	2631
10	2641
20	2646
30	2657
40	2668
50	2682

Temp. degrees F.	Velocity
60	2700
70	2722
80	2750
90	2784
100	2827

Five general equations are commonly used in the development of interior ballistics theory. These are:

1. Equation of state of propellant gases

2. Equation of energy

3. Equation or motion

4. Burning rate equation

5. Form function

The equation of state is used to develop the equation of energy, which in turn is a statement of how the energy released by propellant combustion is distributed during weapon operation. The equation of motion is the solution of forces due to the gas pressure accelerating the projectile.

The burning rate equation determines the rate at which new gas is being generated in the gun by the combustion in the charge. This rate is a function of the pressure of burning, and the area of the reacting surface. If this surface is not constant, it is necessary to introduce the form function to account for effect of the charging burning surface on the rate of generation of gas in the gun.

A detailed discussion of each equation of energy is given in "Interior Ballistics of Guns", AMC Pamphlet #706-150, dated Feb. 1965, Chapter 2.

It should be pointed out that a general theory of gun barrel erosion has not been formulated because the erosion rate decreases as the gun is used due to changes in the interior ballistics resulting from the erosion. A number of general theories have been evolved one of which assumes that, due to roughness, the surface melts only locally, so that erosion is at "hot spots", that shift about on the surface.

Black Powder

Black powder was commonly used as the sole propellant in guns up to the end of the 19th centry, when they were replaced by the nitrocellulose powders.

Being the original "gunpowder" since the 13th centry, it is a mixture of 75% sodium nitrate (saltpeter) 15% charcoal, and 10% sulphur. It has consistently proved to be an undesirable propellant for the following reasons:

1. It burns incompletely, leaving large amounts of residue in the bore.

2. It creates high temperatures locally when burning, causing rapid erosion of the bore.

3. It creates large billows of black smoke.

4. It tends to detonate, developing a high reaction speed that is incontrollable.

5. It must be stored in airtight containers, since it deteriorates when exposed to moisture. Deterioration causes unstable burning.

6. It is highly responsive to friction, shock, and sparks.

7. Black powder dust is highly dangerous.

However, it is used as either a primer or igniter for boosting propellant charges.

Rifling Twist

In specifying rifling twist, the most common practice is to follow prior art, either military or commercial for the same projectile form and weight. Increasing projectile weight causes the projectile to tend toward being unstable unless the twist is sharp enough. Therefore, the twist required varies with the projectile weight.

The torque developed by the rifling in providing rotational acceleration is given by the formula:

$$\text{Torque} = Fd/2 = 2\,AX\pi^2\,pd/NW$$

F = Rotating force

d = Caliber

A_X = Axial moment of inertia (grains-in^2)

p = pressure

N = Twist (calibers/turn)

W = Bullet weight

$A_x = cWd^2$

C = .11 for conventional projectile accordingly, $F = \frac{4cd^2\pi^2 p}{N}$

Gain twist is employed to gradually accelerate the projectile to the required spin rate. The gain twist exit angle is equal to the helical angle of conventional straight twist required for a comparable projectile.

The projectile jacket is a composition of 90% copper and 10% zinc.

Distribution of Energy

For a typical conventional weapon, the energy developed by the propellant, assuming complete combustion, may be distributed generally as follows:

Projectile forward motion	32.00%
Projectile rotation	.14
Projectile friction	2.17
Weapon recoil	.12
Propellant gas motion	3.14
Heat loss to gun and projectile	20.17
Latent heat losses in propellant gases	42.26
Propellant potential	100.00

One particular example, as measured in the Cal. .30 BAR, is as follows, expressed in terms of calories:

Heat to cartridge case	131.	calories
Kinetic Energy of Bullet	885.3	"
Kinetic Energy of Gases	569.1	"
Heat to Barrel	679.9	"
Heat in Gases	598.6	"
Total	2863.9	calories
Frictional Heat Loss	212.0	"

Exterior Ballistics

Exterior ballistics is the study of projectile motion from the muzzle to the target. An understanding of this topic is important to a weapon designer because he should try to promote efficiency in projectile design and effecitveness to the limit of his capacity. That is, a gun designer develops a given weapon to the specifications of minimum weight, maximum reliability and endurance, producing a given velocity to a projectile of a given weight, size, and shape. If the projectile design is of maximum efficiency, the projectile remaining velocity will be high (minimum drag or retardation) and the effective range will be maximized. A gun designer should strive for the maximum effective mileage and penetration at long range of each round fired from his weapon. If the projectile design were lowered in efficiency, then the effective range would be reduced, or effectiveness at the target lowered. Then, in order to increase weapon system effectiveness, a heavier or higher velocity projectile would be specified, resulting in higher pressures, loads, barrel wear, and stresses on the weapon and mount components. Therefore, again, it is most important that close co-ordination be maintained between weapon and ammunition design agencies, as their product is so inter-related, for maximum system effectiveness.

Two important elements of exterior ballistics data are the trajectory and the remaining velocity. The trajectory is the curved path the projectile follows in the air and is a function of gravity and (1) angle of departure, (2) air resistance, (3) shape of the projectile, (4) projectile diameter, and (5) projectile weight. Remaining velocity, of course, is a measure of the projectile velocity at any given range as a function of muzzle velocity and the elements of air resistance that slow the projectile. The remaining projectile energy is then computed, as well as any other function of velocity that effects the lethality of the projectile, as well as other terminal effects.

The study of projectile lethality, or striking energy, is called terminal ballistics, and is a complex study, if all the elements of wound ballistics are to be understood. That is, there is a variety of phenomena that occur when a bullet strikes, and this is a function of projectile velocity, shape, structure, angle of impact, etc., as well as target structure, density, etc., etc. Many of the facets of this subject are not well understood.

The calculation of a trajectory in a vacuum is a common practice in high school physics, but for an accurate treatment, a series of exterior ballistic charts have been prepared that enable anyone to plot trajectories for any cartridge with sufficient accuracy. A typical chart is "Ingalls Ballistic Tables", prepared by personnel of E.I. DuPont De Nemours &Co., Inc. and another is the Speer Ballistic Calculator, slide-rule type of chart. The data is arranged in a series of logarithmic curves (as a slide rule) in order to facilitate the math process.

With these aids, curves of projectile velocity and energy up to 1000 yards can be obtained and compared with each other. It will become apparent how important the "Ballistic Coefficient" is in reducing drag, or velocity retardation, and thus getting more mileage out of the round fired. The ballistic coefficient is, in effect, a measure of the efficiency of the projectile form. The higher the Ballistic coeeficient, ("C"), the better, and vice versa. The value of "C" for the 7.62mm 150 grain boat-tailed projectile is in the order of .4 - .387.

Other common values of "C" are:

Projectile	"C"
.22 Long rifle, 40 gr.	.137
.224-55 gr. SP (flat base)	.209
.223-55 gr. boat-tail	.280
.270 WIN-130 gr. expanding pt.	.496
30/06 Spfld - 110 gr.	.237
30/06 Spfld - 150 gr.	.323
30/06 Spfld - 180 gr.	.560
7.62mm - 125 gr - M43 (Soviet)	.31
.45/70 W.C.F. - 405 gr.	.219

In general, a bullet with a long, smooth ogive, minimum point diameter, boat-tailed, and high in weight, will have a good ballistic coefficient. The long ogive radius reduces head pressure, and the boat-tail reduces suction at the base.

The ballistic coefficient is calculated as $C = W/id^2$, where

w = projectile weight in lb.

d = projectile diameter in inches

i = coefficient of form.

The coefficient of form depends upon the ratio of the bullet ogive to the bullet diameter and the effect of air resistance on the point. Bullets of different calibers that have the same shape (mere scale-ups) have the same coefficient of form. The tables given in the DuPont charts have been compiled from firing, as well as tabular, data, from several sources; being averaged whenever there were differences.

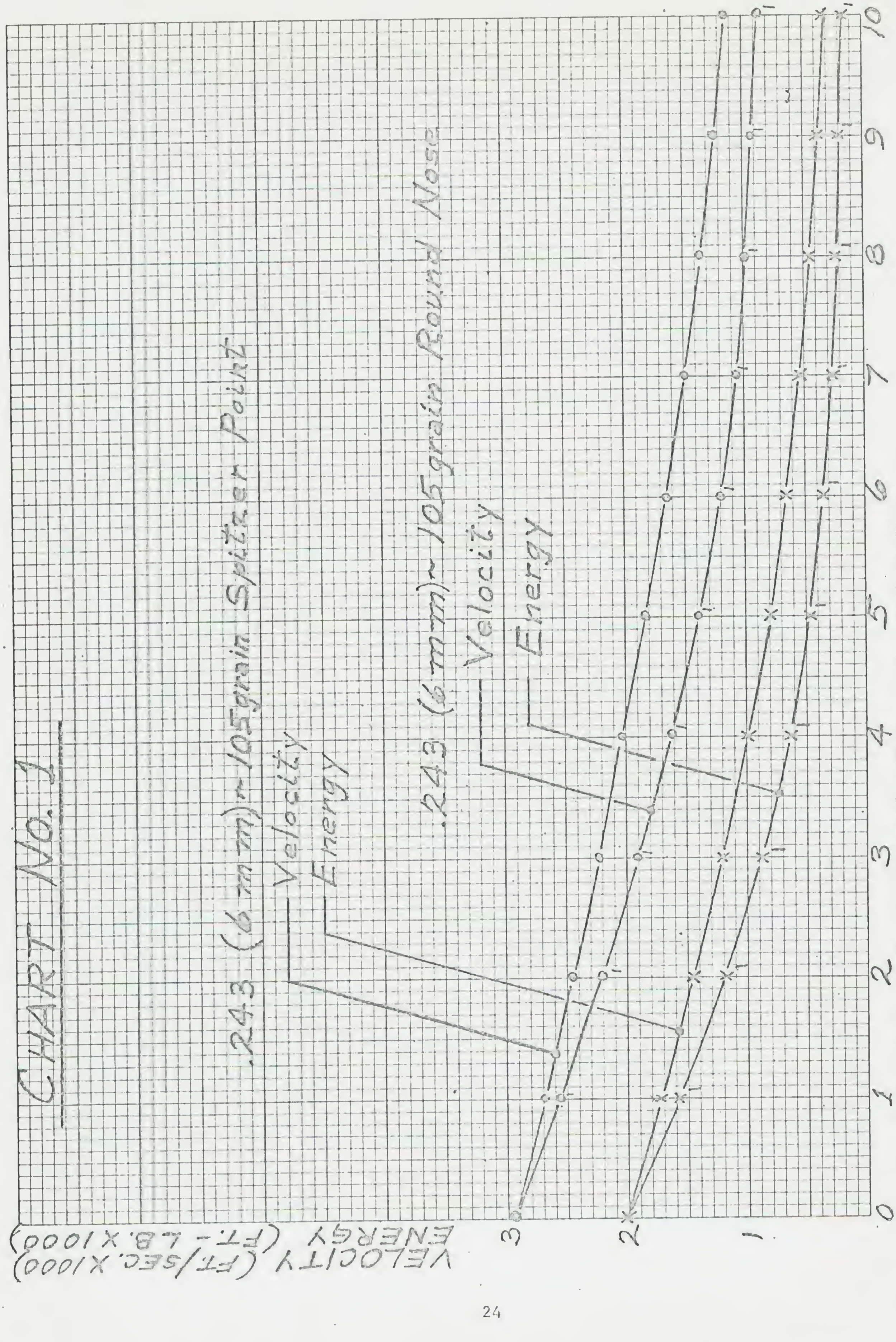


CHART NO. 1
.243 (6 mm) ~ 105 grain Spitzer Point
Velocity
Energy
.243 (6 mm) ~ 105 grain Round Nose
Velocity
Energy
VELOCITY (FT./SEC. X 1000)
ENERGY (FT.-LB. X 1000)
RANGE IN HUNDREDS OF YARDS
0
1
2
3
4
5
6
7
8
9
10

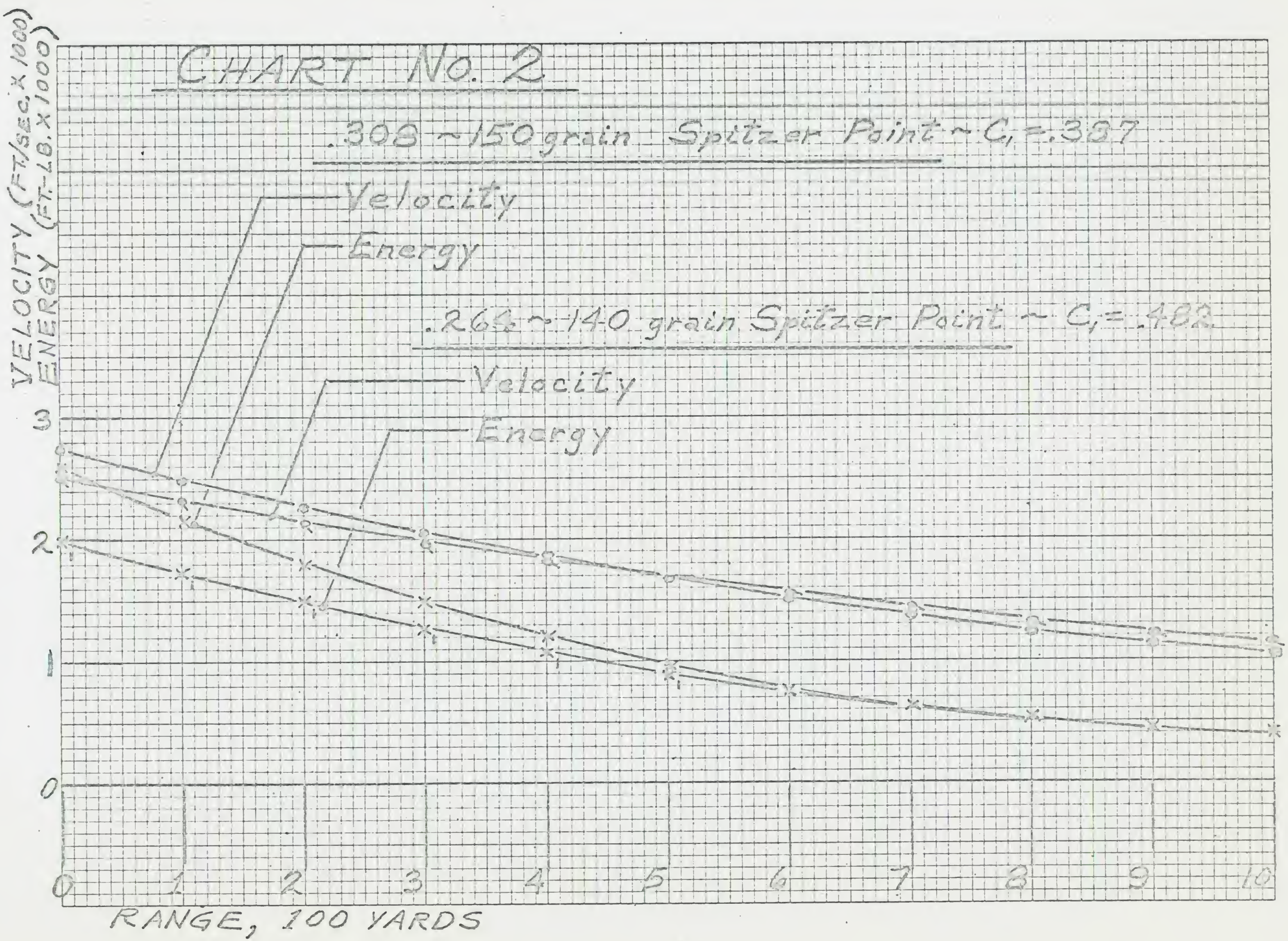


CHART No. 2
.308 ~ 150 grain Spitzer Point ~ C1 = .337
Velocity
Energy
.264 ~ 140 grain Spitzer Point ~ C1 = .482
Velocity
Energy
VELOCITY (FT/SEC. X 1000)
ENERGY (FT-LB. X 1000)
3
2
1
0
0
1
2
3
4
5
6
7
8
9
10
RANGE, 100 YARDS

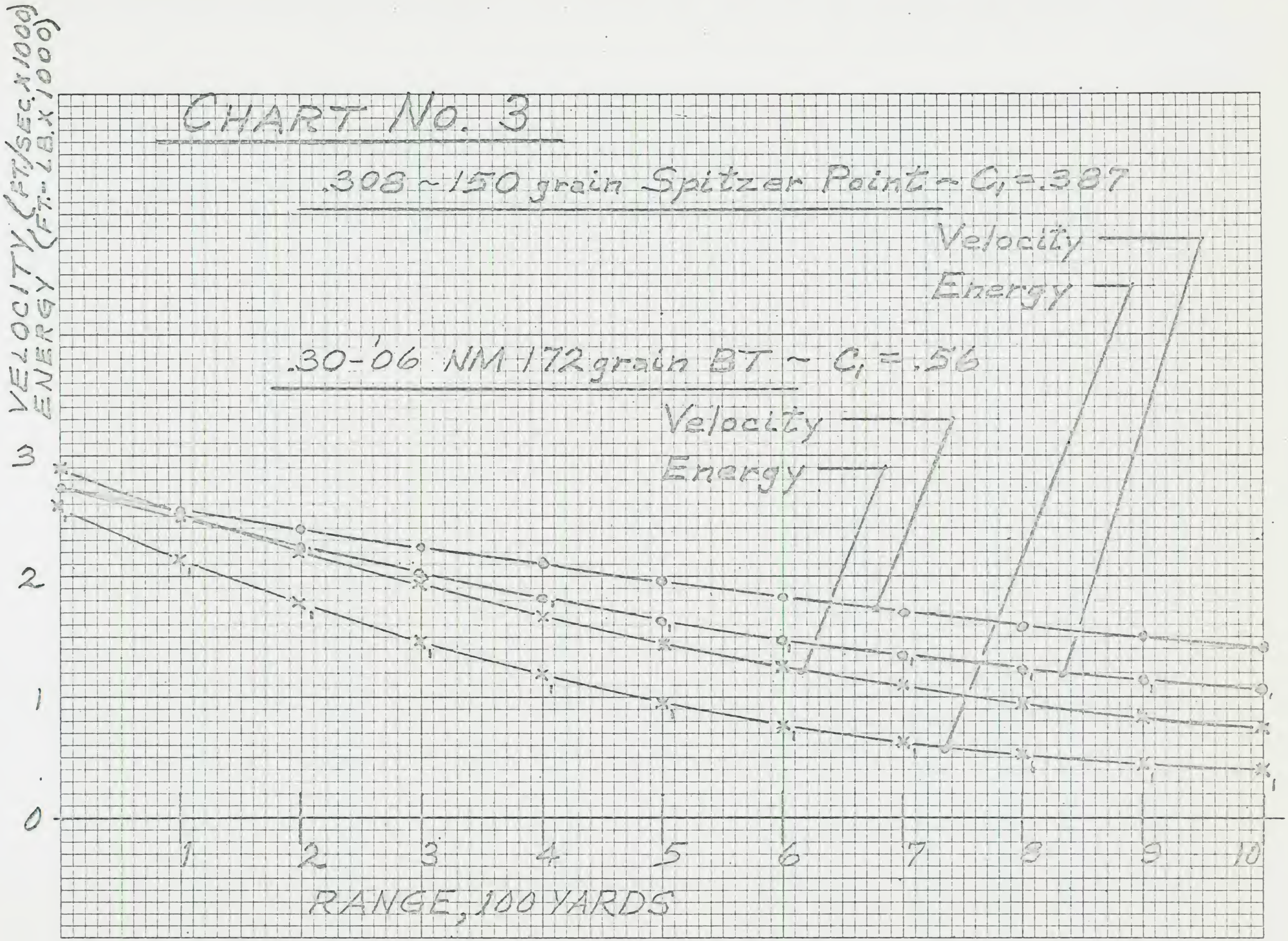


CHART No. 3
.308 ~ 150 grain Spitzer Point ~ C_1 = .387
Velocity
Energy
.30-06 NM 172 grain BT ~ C_1 = .56
Velocity
Energy
VELOCITY (FT/SEC. x 1000)
ENERGY (FT.-LB. x 1000)
3
2
1
0
1
2
3
4
5
6
7
8
9
10
RANGE, 100 YARDS

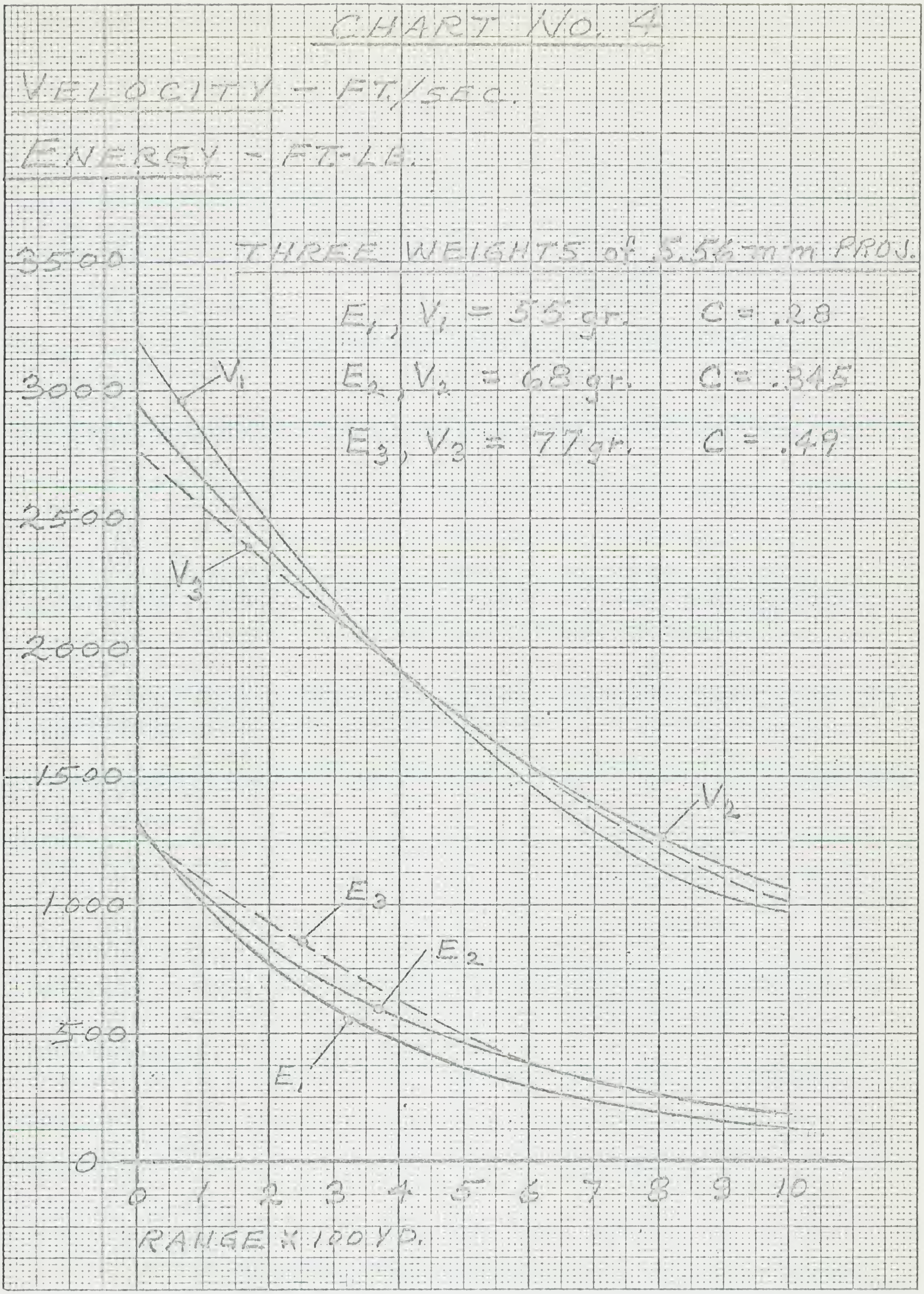


CHART No. 4
VELOCITY - FT./SEC.
ENERGY - FT-LB.
THREE WEIGHTS of 5.56 mm PROJ.
E1, V1 = 55 gr. C = .28
E2, V2 = 68 gr. C = .315
E3, V3 = 77 gr. C = .49
3500
3000
2500
2000
1500
1000
500
0
V1
V2
V3
E1
E2
E3
0 1 2 3 4 5 6 7 8 9 10
RANGE x 100 YD.

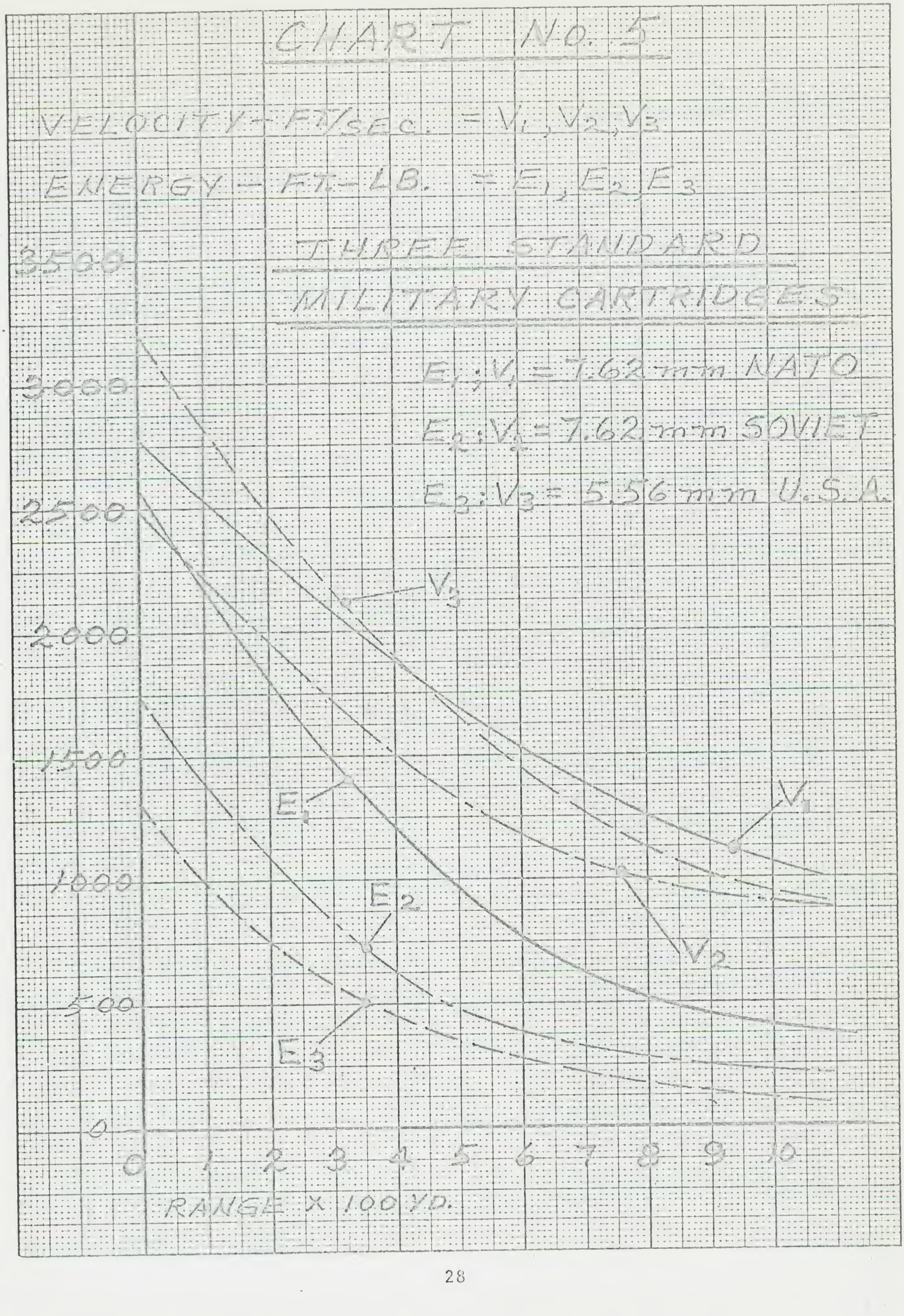


CHART NO. 5
VELOCITY – FT/SEC. = V_1, V_2, V_3
ENERGY – FT.–LB. = E_1, E_2, E_3
THREE STANDARD
MILITARY CARTRIDGES
$E_1 ; V_1$ = 7.62 mm NATO
$E_2 : V_2$ = 7.62 mm SOVIET
$E_3 : V_3$ = 5.56 mm U.S.A.
3500
3000
2500
2000
1500
1000
500
0
V_3
V_1
V_2
E_1
E_2
E_3
0 1 2 3 4 5 6 7 8 9 10
RANGE X 100 YD.

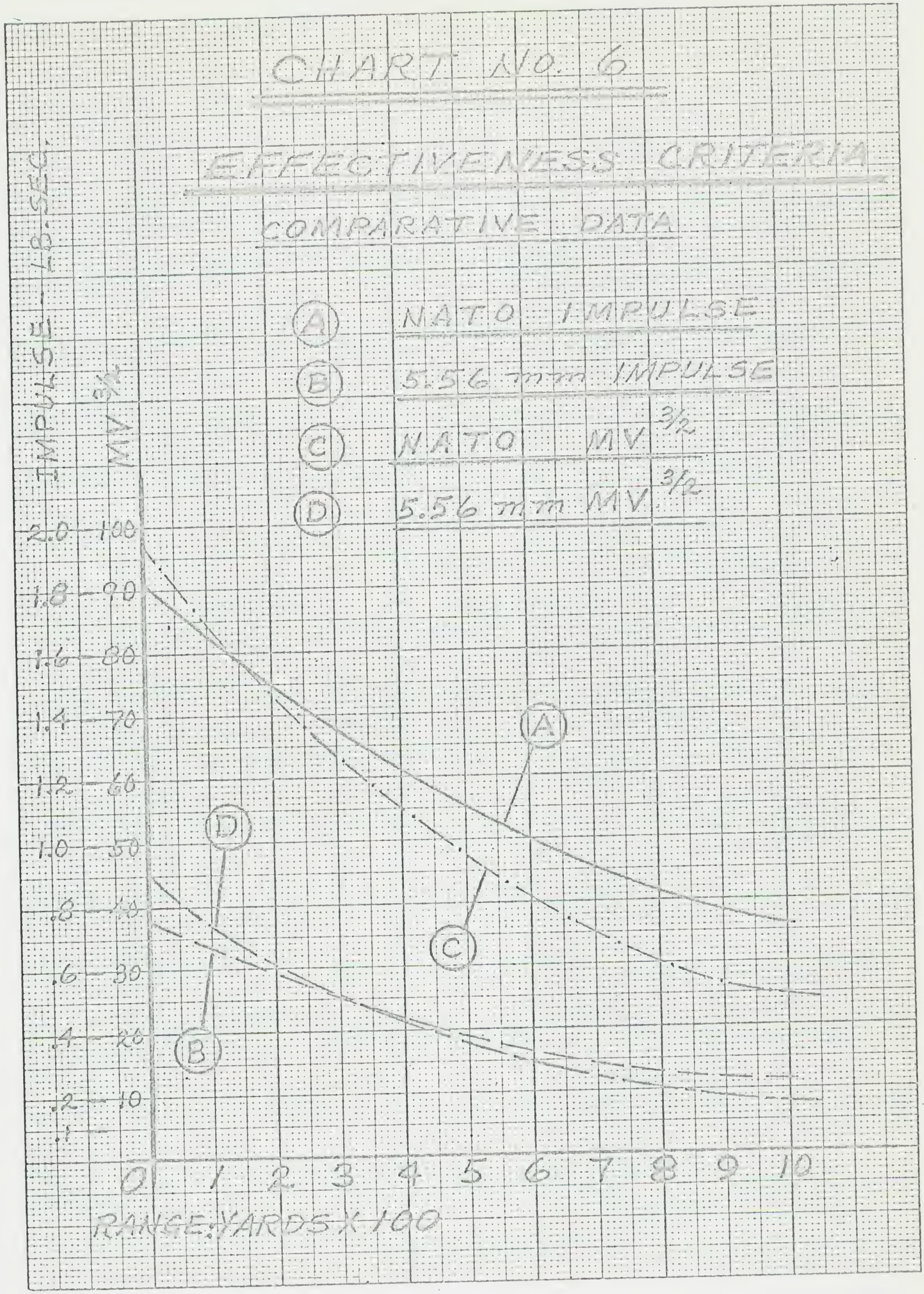


CHART No. 6
EFFECTIVENESS CRITERIA
COMPARATIVE DATA
(A) NATO IMPULSE
(B) 5.56 mm IMPULSE
(C) NATO MV 3/2
(D) 5.56 mm MV 3/2
IMPULSE - LB. SEC.
MV 3/2
2.0 100
1.8 90
1.6 80
1.4 70
1.2 60
1.0 50
.8 40
.6 30
.4 20
.2 10
.1
0 1 2 3 4 5 6 7 8 9 10
RANGE: YARDS X 100
A
B
C
D

In using the DuPont chart for bullet form, match the bullet profile carefully, noting that the sharp points on the diagram are for drafting reference only. The table makes adjustments for the diameter of the hollow point or flat nose.

The charts use a series of vertical reference planes and scales for each of the factors in the equation. Following the step-by-step directions for the given charts, data may be obtained for plotting the curve of remaining velocity up to 1000 yards range; the angle of departure; the time of flight; the maximum height of the trajectory; the angle of fall; wind deflection; and the remaining energy. The charts on the following pages compare the remaining velocities and energies of a number of commercial and military rounds. Note that the comparison of values at the muzzle is not the same as the comparison at, say, 500 or more yards. This is where the ballistic coefficient makes a significant difference.

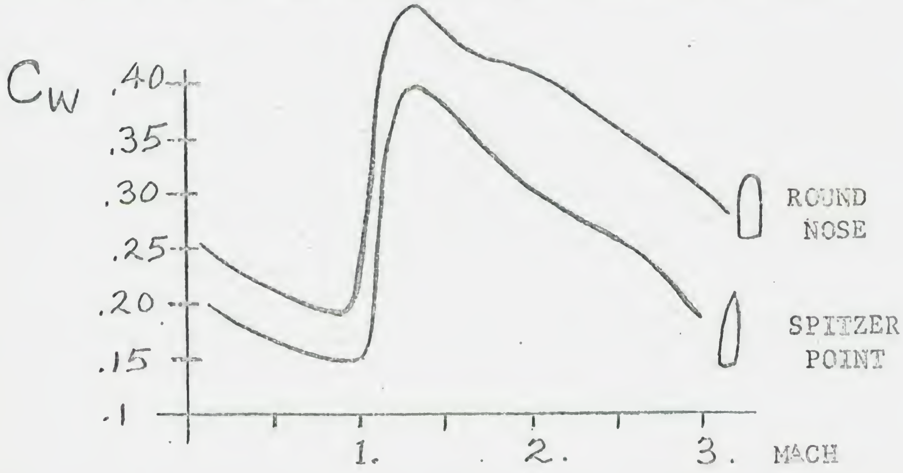
Charts #1, 2, and 3 are related to a following dissertation on "Ballistic Coefficient".

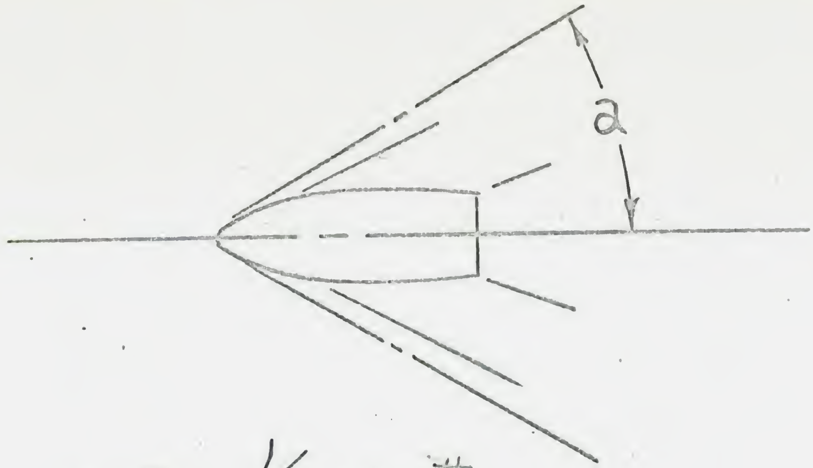
Chart #4 shows the effect of increasing the weight of, for example, a 5.56mm projectile. The resultant change in rate of velocity loss reflects a variation in the remaining energy level over the range of 1000 yards. In chart #5, values of remaining velocity and energy over the range of 1000 yards are shown for the three current military cartridges.

In addition, when the remaining velocities are determined by the use of ballistic tables, a variety of data may be derived, as shown in chart #6. Here, two philosophies of projectile effectiveness are shown in a comparison of the 7.62mm NATO and the 55 grain 5.56mm cartridge.

Remaining impulse and MV 3/2 curves reflect a comparison of these two standard cartridges over the range of 1000 yards.

The coefficient of air resistance as a function of the Mach number is shown as follows: (p. 181 Oerlikon handbook)





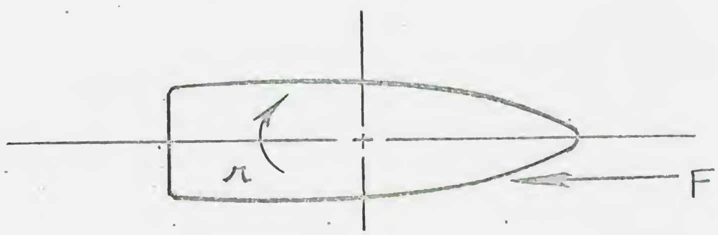
$$\sin a = \frac{1}{\text{MACH}^{\#}}$$

Spark photogaphic techniques of a projectile in flight permit measurement of the velocity as a function of the Mach number.

The complex nature of the bullet's trajectory is due to the fact that for a spin stabilized projectile, the mass of the projectile is not concentrated at the center of gravity and the projectile does not travel exactly in the direction of its axis, but yaws, oscillating about the tangent of the trajectory. This angle is the angle of precession, and is caused by the air resistance and moment with respect to the center of gravity.

The projectile in flight is stabilized as a gyroscope is; by spin. Such a rotating body possesses an angular momentum that tends to resist forces that act to tilt the spin axis.

In comparing the behavior of a gyroscope, if a force is applied on the head of the projectile, F,



the projectile will turn, not about the horizontal axis, but will precess, or turn, about the vertical axis. This motion is called precession, and is due to the spin imparted by the rifling.

The force causing bullet deflection is not a lateral force against the side of the bullet, but a force applied at some other point on the ogive. There, it will tend to tip the bullet, and gyroscopic precession must result.

With comparable rates of spin, the angular momentum of a short fat bullet is much greater than that of a long thin one of equal weight, and the short fat one accordingly will precess more slowly and deflect less. Thus it will result in less deflection from branches and twigs. This is why round nose bullets have better brush - bucking ability than spitzer - point bullets.

Another familiar phenomenon is called bullet drift. It is well known that bullets tend to curve to the right of the line of sight when fired over long range, assuming a right hand rifling twist. This is also due to the gyroscopic effect. The force opposing gravity, on the ogive area forward of the center of gravity, causes precession to the right, because of the right hand spin. The yaw is the attitude of the projectile with respect to the longitudinal axis.

Likewise, windage will cause the projectile to strike higher or lower, depending upon wind direction. Wind blowing from right to left will cause bullet drift to the left, and at the same time (assuming right hand twist) due to gyroscopic precession, will cause the bullet to strike low.

Wind direction from left to right will conversely cause the bullet to drift to the right, and strike high.

The following informal dissertation on "The Ballistic Coefficient", by this author, again cutlines the significant characteristics of the ballistic coefficient. Of particular importance is the comparison of the .264 (6.5mm) and the 7.62mm NATO remaining energies, which are equal from 600 yards to 1000 yards. (Chart #2).

Also significant is the comparison between the NATO round and the Cal. .30 - 172 grain bullet which shows, for example, that the heavier bullet has as much energy at 1000 yards as the NATO has at 650 yards. This is where firepower is improved without paying a penalty in the weapon.

Ballistic Coefficient

What is it? Very simply, it is a measure of the efficiency of a projectile in flight through the air. If we were firing in a true vacuum, there would be no need to consider a ballistic coefficient,

but firing in a true vacuum only occurs in high school textbooks, and the rest of us have to put up with air resistance, which creates frictional resistances and drag. This causes the projectile to steadily lose velocity. The rate at which velocity drops is determined by the Ballistic Coefficient. The higher the value of Ballistic Coefficient, the greater the efficiency of the projectile in flight.

The practical value of studying the Ballistic Coefficient should be apparent to those who seek a flatter trajectory, more impact energy at longer range, and a longer effective range. Instead of cramming more or hotter powder into the chamber to increase energy, which, by the way, adds dangerously higher strains to the barrel, locking lugs, and cartridge case, and shooter should pay closer attention to the bullet weight and shape. These are the factors that determine Ballistic Coefficient. The most efficient projectile would be essentially heavy (some even consider the use of a depleted uranium core) for maximum density, but retain a long smooth ogive (Spitzer point) with minimum flat point, and a boat-tail, to reduce the aft drag, or turbulence. The ideal bullet tip would be pointed, but this is impractical for handling in the magazine, therefore the flat tip diameter should be about one tenth of the caliber.

Other good reasons for paying attention to bullet design for increased effectiveness at longer range, rather than making the chamber hotter, are that, first with increased load, the recoil impulse is higher, loading your shoulder all the more, which also will affect your accuracy, and that higher loads reduce the number of times you can safely reload your cartridge. Also, barrel wear is reduced and the wear and tear on automatic mechanisms is also to be considered.

There are three factors in the standard formula for computing the Ballistic Coefficient and these are weight, bullet diameter, and form factor. The bullet weight divided by the bullet diameter squared is a measure of sectional density, and the higher this is, the better your Ballistic Coefficient. The form factor is a value based on the shape of the projectile. The longer the ogive radius and the smaller the tip flat diameter, the better the form factor. Boat-tailing the projectile also improves form factor.

Now let's look at some practical results of this Ballistic Coefficient in action. For example, study the curves shown in Chart #1, which plot velocity and energy of two different bullets out to 1000 yards. Each bullet is a standard 105 grain 6mm (cal..243) sample with a muzzle velocity of 2964 feet per second. The difference is that one is a round nose bullet with a Ballistic Coefficient of .256, while the other is a spitzer point bullet with a Ballistic Coefficient of .395. Of course, since bullet weights, diameters and muzzle velocities are identical, there is no difference in travel in the bore, since the energy from the expanding gases is identical in each case. After the projectiles

leave the muzzle and are free in the air, then the difference in Ballistic Coefficient shows up markedly. Immediately the respective velocity and energy curves begin to separate from their common starting point at the bore, so that eventually, the spitzer point has the same energy at 1000 yards that the round point bullet has at 650 yards. Thus, by picking the spitzer point, you gain 350 yards in range effectiveness. Likewise, with velocity, for example at 600 - 700 yards, the spitzer bullet velocity is 40% greater than the round nose bullet.

This is not a condemnation of round nosed bullets, however. They have their place. The principle advantage of round nose bullets is their stability in brush bucking, when compared to Spitzer point bullets. Therefore, if you are hunting in wooded or brushy terrain, where you target is likely to be at medium or shorter ranges, use round nose bullets. If your hunting range is open country and the target is likely to pop up at longer ranges, then the Spitzer point bullet is recommended.

The criteria for effectiveness shown here is energy. Effects in tumbling and other wound ballistic phenomenon depend upon the portion of the game hit, whether boney, fluid, effectiveness against vital organs, etc., and are so variable that many arguments on this subject have burned the midnight oil, each hunter having a different opinion which naturally is based on his experience and observations. The subject therefore need not be exhausted further in this discussion.

Another interesting effect of an improved ballistic coefficient for a number of projectiles is that their performance is upgraded to the level of higher caliber cartridges at longer ranges. For example, note Chart #2, which compares a (.264) 6.5mm 140 grain bullet having a muzzle velocity of 2500 fps and Ballistic Coefficient of .482 with a (.308) 7.62mm 150 grain bullet having a muzzle velocity of 2740 fps and Ballistic Coefficient of .387. Both bullets are spitzer points. The big difference in Ballistic Coefficient is due to the smaller diameter of the 6.5mm coupled with its weight of 140 grains. This is known as sectional density, which is one reason why lead cores are used.

Looking at the curves, you will see that the 6.5mm muzzle energy is 2000 ft.-lb., while the 7.62 muzzle energy is 2600 ft.-lb. However, because of improved sectional density, or Ballistic Coefficient of the 6.5mm, the energies are equal at about 600 yards. That is, the 6.5mm caught up to the 7.62mm at that range, and equalled it on out to 1100 yards, at least. In fact, at those longer ranges, the 6.5mm remaining velocity is higher than the 7.62mm.

Many of you may recognize the 7.62mm example as the U.S. standard NATO military cartridge. This is true, and points out how the NATO Cartridge may be improved by increasing its Ballistic Coefficient. In fact, this can be done quite simply by substituting another standard 7.62mm projectile, the 172 grain boat-tailed National Match bullet,

boasting an impressive Ballistic Coefficient of .56. Of course, a slight adjustment in powder charge is made in order to maintain peak pressure at the same level. That is, increasing bullet weight (without changing powder charge) causes the peak pressure to rise in the chamber. This is because the expanding gases are pushing against a heavier bullet, causing a slightly slower acceleration, or take-off, of the bullet; therefore, if you can imagine you are keeping the lid tighter on a boiling pot, the gas pressure will be higher, therefore either a slower burning powder is used, or a few grains of propellant are lopped off.

Chart #3 shows the vast difference in remaining energies for the two 7.62mm projectiles. For example, the 172 grain N M bullet has the same energy at 1000 yards that the 150 grain NATO has at 600 yards, and the NM energy at 1000 yards is double the NATO energy at 1000 yards. This is particularly advantageous when shooting in open country.

As a curious sidelight, compare the Ballistic Coefficient of the NATO (150 gr.) bullet with the old-time lead ball. The ball, being a sphere, will have a poor form factor, about 1.4 (the Nato is .6), and a corresponding Ballistic Coefficient of .152 times the ball diameter (which makes it easy to compute for all size lead balls). For a .30 cal. ball, the Ballistic Coefficient then is .0455 while the NATO is .387, even though the lead ball will be heavier, weighing in at about 200 grains (NATO, 150 grains).

The obvious result is that the lead ball, while potent at the muzzle, will lose energy very quickly. This is the main reason that shot pellets have such a short effective range of only 60-80 yards.

In summary, then, selection of a more efficiently streamlined bullet, (better surface finish, too, when casting bullets) will pay off more handsomely in striking energy than loading the case with a hotter powder charge. The benefits will also include a shorter time to target, a flatter trajectory, and, therefore, improved accuracy.

Recoil

"Recoil" is nothing more than an expression of Newton's third law, which states that "for every action there is an equal and opposite reaction". The determination of recoil values is fundamental in ordnance, as it represents a measure of the force produced for each round fired.

The value of "free recoil" is usually considered in calculations of recoil energy. That is, not considering additional masses or resistances such as the shoulder or forearm that, in reality, make recoil "effect" less violent than the "free recoil" calculated. The calculation of free recoil is useful for purposes of comparison of one weapon, or charge, against another.

In fact, and as bourne out by experience, recoil effect or "kick" is lessened by holding the butt firmly against the shoulder. This affects the acceleration of the skeletal frame. Thus the resulting recoil energy is more "enjoyable" if it is a "push" rather than a "blow".

Some authorities place a limit on maximum recoil energy for a military rifle of about 15 foot-pounds. A person can briefly handle double that amount, but the frequency is not expected to approach that which the infantryman will fire. As an extreme, a fifteen pound elephant rifle may commonly have a recoil energy in excess of 50 foot pounds.

Typical recoil energies of several standard weapons are calculated as follows:

a. Recoil energy of Cal. .30 M1 rifle:

Projectile Impulse = 2.3 lb.-sec.

Rifle impulse then = 2.3 = WV/g

$V = 2.3 \times 32.2/9 = 8.25$ ft./sec.

$E = WV^2/2g = 9 \times 68/64.4 = 9.5$ ft. lb.

b. Grenade Launcher M79:

Projectile impulse = $.375 \times 240/32.2$
= 2.8 lb.-sec.

Recoil Velocity = $2.8 \times 32.2/6.5 = Ig/W$
= 14 ft./sec.

Recoil Energy = $WV^2/2g = 6.5 \times 194/64.4$
= 19.6 ft.-lb.

Often the question is broached as to the amount of rifle motion prior to bullet exit. Taking the common 30'06 rifle as an example, firing a 180 grain match bullet, popular in competitive target shooting in a 24 inch barrel, the solution is easily found. Simply consider the distance the bullet travels as being inversely proportional to the rifle recoil travel as the weights of bullet and rifle.

Consider 1/2 of the propellant charge as traveling with the bullet, since this is true up to the point of muzzle exit.

$$\frac{Sr}{Sp} = \frac{Wp}{Wr}$$

$$\frac{Sr}{24} = \frac{(180 + 25)/7000}{9}$$

$$\underline{Sr = .078 \text{ inch}}$$

Various methods of measuring recoil are used, the preferred method being to hang the gun on parallel wires, and measuring the velocity of recoil of high speed cameras or other instruments.

This eliminates the use of secondary calculation, which do not correct for certain losses.

In another method, a steel ball is attached to the buttplate and the indentation made in a lead block is measured. (Ala rockwell hardness testing.) This method is good for comparative measurements only.

To calculate recoil data, the basic law of course, is F = m a. Since a = v/t then F t = mv. The term "m v" is referred to as the "momentum", or the property of a given mass at a given velocity, while "F t" is the "impulse", or the effect of a given force acting for a given time.

For recoil forces, impulse is equivalent to the average force that would bring the moving body to rest in one second.

There are three vectors contributing to recoil and they are:

(1) Reaction to projectile acceleration

(2) Reaction to propellant motion

(3) Muzzle blast, both gas exit and action on muzzle face.

The first factor is the most important, and while the impulse of the projectile equals the weapon impulse, the energy is a function of velocity squared, therefore, the weapon energy is usually only about 1/200 of the projectile energy.

The effect of the muzzle blast is a function of muzzle pressure. Therefore, shortening the barrel does not reduce recoil, as one may think, since the muzzle velocity is lowered. Rather, recoil will be intensified because of the much higher muzzle pressure, usually quite noticeable. This is equivalent to a rocket thrust. and is determined by the equation:

Thrust = Muzzle pressure X Area X Mass rate of discharge. (This factor is a function of propellant chemistry).

An approximation that compensates for the unknown in muzzle blast equations is the practice of assigning a velocity of 4700 for gas exit velocity. This agrees with dynamometer tests for military small arms.

Thus the formula for recoil velocity of the weapon is approximately $Vw = (WpVp + 4700C)/Ww$

The energy then, $E = \frac{WV^2}{2g}$

The value of weapon impulse in firing is best determined by test firings in Ballistic pendulums or by using a Velocimeter to determine velocity of muzzle gases. It is this variable that limits the accuracy of conventional analysis. For example, the stated formula using a charge velocity of 4700 fps at the exit, is restricted to small arms with a projectile velocity in the range of 2400 - 2800 fps class. Further, the weapon impulse formula may be divided into two phases: First, impulse during projectile travel to the muzzle, and Second, impulse at muzzle exit.

For the first phase, the weapon impulse equals the projectile impulse plus the powder impulse at one half the projectile muzzle velocity, since the powder expands at one half the rate of the projectile velocity.

This formula, then, is:

$I = Ww\,Vw/g = WpVp/g + 1/2\ WcVp/g$

or, $Vw = Vp\ (Wp + .5Wc)\ /Ww$

After muzzle exit, additional impulse is realized due to the gas blast.

Extensive experiments indicate that this can be compensated for by increasing the effective mass of the powder charge as follows:

(1) For Shotguns and Revolvers with long barrels:

$$V_w W_w = V_p (W_p + 1.25 W_c)$$

(2) Krag Rifles and shotguns and revolvers with short barrels:

$$V_w W_w = V_p (W_p + 1.5 W_c)$$

(3) M1903 Rifle & M1 Rifle:

$$V_w W_w = V_p (W_p + 1.75 W_c)$$

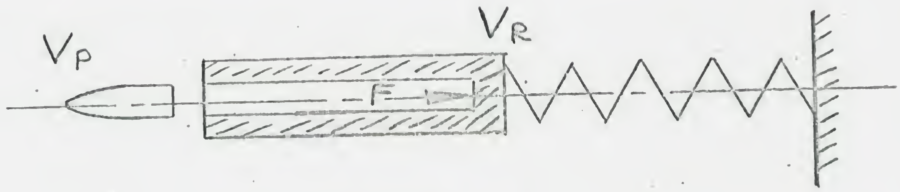
(This is comparable to assigning a charge velocity of 4700 at <u>full charge weight</u>, rather than 1/2 charge)

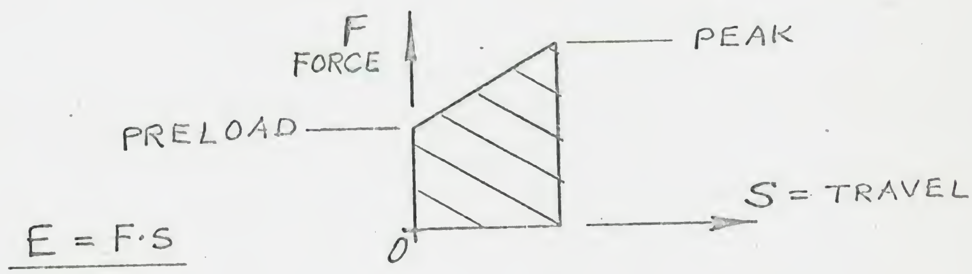
Pendulum data taken at Springfield Armory show an impulse of 2.51 lb.-sec. for the M14 rifle. This resulted in a constant of 1.05 ($V = 2860$). (Impulse varied from 2.45 - 2.55)

The mount, or gun support, must react to the recoil energy, and the value of this force depends on the elasticity of the mounting.

The conditions of loading for resilient mounts depend upon the stiffness of the spring; for stiff springs the load is high, while recoil is short, while for softer springs, the load is lower, but a longer permissible recoil travel is required. The weapon rate of fire requirements determine spring rates.

In the following schematic, note that the area under the spring load diagram approximately equals the recoil energy:

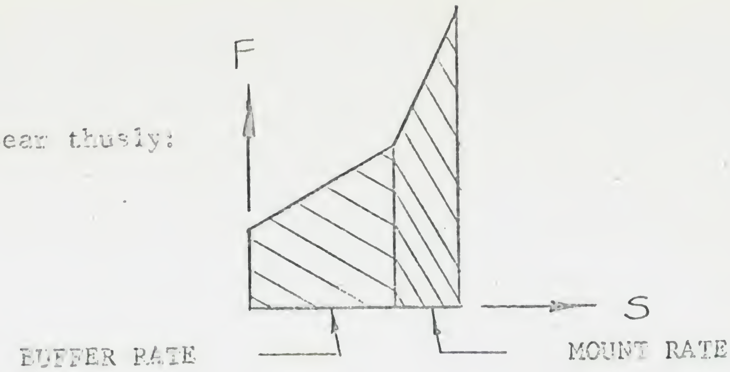




Ordinarily recoil systems will have secondary loading diagrams, which may be due to buffers, resilient pads, or the frame work itself.

The diagram would appear thusly:

$$E = \int F \cdot S$$



Therefore, in laying out spring loads for assembled and minimum heights the area under the trapezoidal diagrams should represent the energy of the recoiling mass.

A study of the recoil effects of shoulder fired small arms weapons was conducted at BRL, Aberdeen, Maryland, and the following observations were made:

a. The free recoil momentum and energy of rifles firing AP ammunition were greater than those firing ball ammunition.

b. The average escape velocity (at the muzzle) of the powder gas was less for AP ammunition than it was for ball ammunition. A greater part of the powder energy was apparently transmitted to the AP projectile.

c. Experimental muzzle brakes produced a considerable decrease in the gas escape velocity, while a cone-type flash hider produced an increase.

d. The recoil momentum of the M1 rifle is slightly less than that of a Springfield rifle, most likely due to the escape of gas at the breech on opening.

In another series of tests, a variety of shooters fired the M1 rifle from a standing position, without use of a sling. Measurements of force vs. time and displacement vs. time were made. Results indicate:

a. The average maximum force and velocity on 10 out of 11 men was greater when firing AP ammunition (over ball ammunition).

b. The variation of force and velocity from man to man was relatively large.

c. The maximum velocities are lower than the free recoil velocities in every case. This is due to the shooter's arm and shoulder weight, as well as his resisting force. The effective shoulder weight varies from 3.5 to 5.0 pounds, and neglects the resisting forces.

d. It appears that the way a rifle is held has a greater effect on recoil than the size or build of a rifleman.

Quantitative values were found as follows:

a. Free recoil measurements (five wire pendulum) of M1 Rifle (weight rifle = 9.72 lb.)

(1) Ball ammunition:

(Projectile)	Wp Vp =	58.63 ft. lb./sec.
(Charge)	Wc Vc =	30.77 ft. lb./sec.
(Rifle)	Wr Vr =	89.40 ft. lb./sec.
	Vc =	4270 fps
	Vr =	9.2 fps
Rifle	KE =	12.8 ft.-lb.

Average Maximum force (11 shooters) = 69.3 lb.
Standard deviation = 3.6 lb.

(2) AP ammunition:

Wp Vp =	64.71 ft. lb./sec.
Wc Vc =	30.07 ft. lb./sec.
Wr Vr =	94.78 ft. lb./sec.
Vc =	4170 fps
Vr =	9.75 fps
KEr =	14.4 ft.-lb.

Average maximum recoil force = 73.7 lb.
Standard deviation = 3.2 lb.
(Note that WpVp + WcVc = WrVr)

Dynamics of Automatic Rifles

The tactical requirements of a full automatic weapon are quite different than those of a semi-automatic rifle, so a weapon design that seeks to satisfy both functions can only be compromise, at best. (M-14 vs BAR). The BAR, being a heavy automatic rifle, can be held on the target area, whereas the lighter M14, with a dropstock, climbs badly in automatic fire.

When a round is fired, an impulse develops both translational and rotational velocities in the gun. This is not a significant problem in semi-automatic fire, but in full-auto, riflemen have difficulty holding the weapon on the target.

The induced rotational velocity, multipled by the time lapse between shots represents the angular deviation between successive shots. This widens for each round because of the initial velocity input for successive rounds. Therefore, an effective automatic fire program should be limited to short bursts if any degree of accuracy is desired.

With "w" representing angular velocity increments per round, "r" the rate of fire, "n" the number of rounds in a given burst and "θ n" the accumulated deviation for the burst, the following formula is applied:

$$\theta\, n = 1/2\, \frac{w}{r}\, n\, (n - 1)$$

Therefore, the deviation varies as the angular velocity increment and inversely as the rate of fire. Therefore, firing rate should be higher for this application. However, if the geometry and weight of the weapon is such that the shooter can be trained to "hold" the rifle on target between rounds, then a lower rate may be desired, as in the BAR.

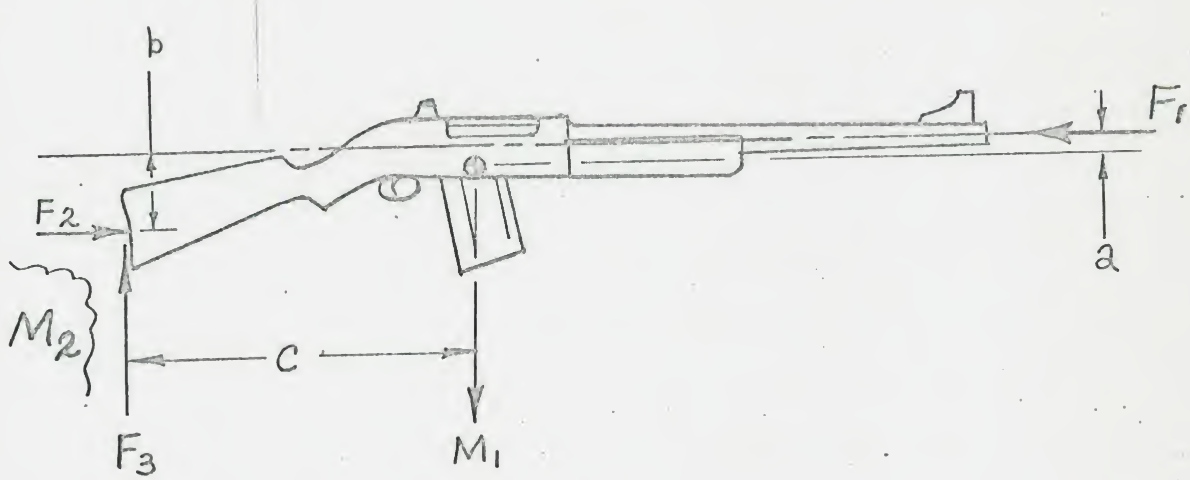
The "salvo" concept is concerned with the "number of rounds" that are fired into a target area before a significant deviation occurs. This is in line with the "shotgun" concept. Note that the formula contains an n^2 factor; therefore, the deviation is accelerating with burst length. As an example, with a firing rate of 600 spm and a "w" per shot of 20 mils per second; the deviation as a function of burst length is as follows:

Shot No.	Deviation (mils)
1	0 (assumed)
2	2
3	6
4	12
5	20
6	30
etc.	

Therefore, a three round burst, as utilized in the SPIW system, would offer short bursts of maximum effectiveness and prevent needless waste of ammunition.

One critical user test is the "quickfire" course, in which the shooter fires at short-range random targets from the hip. Here, rifle profile is important, as noted in tests with the M16 rifle. With the shooter's eye approximately 18 inches above the weapon, the shooter's projected sight line runs from the top of the carrying handle to the muzzle. This creates an optical illusion that cause the bore to be projected at a sharp upward angle, causing the shooter to hit high.

DYNAMICS of AUTOMATIC RIFLES (Typical)



⊕ = Center of Gravity

RESULTANT WORKING FORMULA:

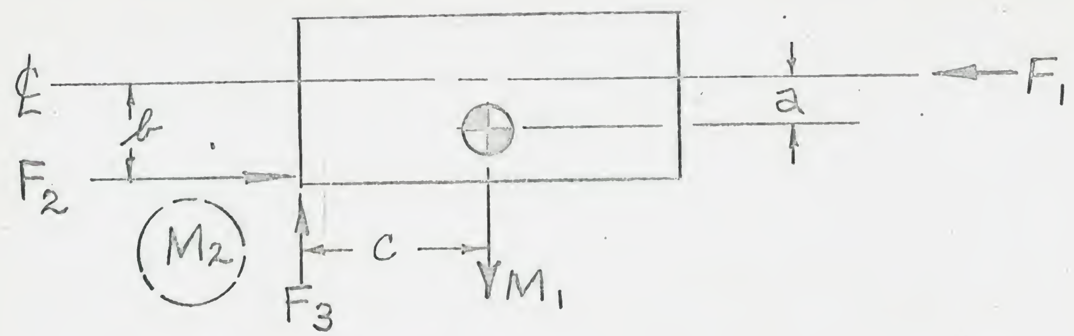
$$\omega = M \left[\frac{M_1 a + M_2 b}{(I + M_1 c^2)(M_1 + M_2) + M_1 M_2 (b - a)^2} \right]$$

ω = angular velocity (radians per second)

I = moment of inertia (ft.-lb.-sec.^{2})

M = Momentum (lb.-sec.)

Consider the following force diagram:



F_1 - Propelling Force

F_2 - Shoulder Reaction (horizontal)

F_3 - Shoulder Reaction (vertical)

a - Bore ℄ to c.g.

b - Bore ℄ to shoulder

c - c.g. to shoulder

$\bar{a}$ - angular acceleration of weapon

w - angular velocity of weapon

I - moment of inertia about c.g.

M_1 - Weapon mass

M_2 - Mass of shooter (equivalent)

M - Momentum of bullet and charge

Dynamics of Automatic Rifles

The moments and reactions about the c.g. are as follows:

$$(1) \quad I\,\bar{a} = F_1\,a + F_2\,(b-a) - F_3\,c$$

The value of M_2 varies greatly and cannot be calculated, but it has been observed that a value of 1/4 the shooter's weight is reasonable. This is an empirical value for the shooter's supporting weight plus the force applied by the shooter to resist recoil. In any event, since it is many times the rifle weight, variations have little practical effect.

The horizontal acceleration of the gun butt is: ($F = m\ a$)

$$(2)\quad \frac{F_2}{M_2} = \frac{F_2 - F_1}{M_1} + (b-a)\ \bar{a}$$

so that

$$(3)\quad F_2 = \frac{M_2\ F_1 - M_1\ M_2\ (b-a)\ \bar{a}}{M_1 + M_2}$$

The vertical acceleration moment is:

$$(4)\quad \frac{F_3}{M_1} - C\ \bar{a} = 0$$

so that

$$(5)\quad F_3 = M_1\ C\ \bar{a}$$

Substituting equations (3) and (5) into (1)

$$I\bar{a} = F_1\ a + \left[\frac{M_2\ F_1 - M_1\ M_2\ (b-a)\ \bar{a}}{M_1 + M_2}\right](b-a) - M_1\ c^2 a$$

$$\bar{a} = F_1 \left[\frac{M_1\ a - M_2\ b}{(I + M_1\ C^2)(M_1 + M_2) + M_1\ M_2\ (b-a)^2}\right] = F_1 \left[\emptyset\right]$$

Since angular velocity is a function of acceleration as M is a function of F_1,

($\int \bar{a}\ dt = w$ and $\int F_1\ dt = M$)

therefore $w = M \left[\emptyset\right]$

(w in radians/sec)

By inspecting this formula, you see that if a and b are near zero, (an in-line condition), w vanishes. Also note that due to the M2/M1 ratio the "w" is more sensitive to the "drop" of the stock than to the positive of the c.g.

However, rifles with drop stock configurations are designed for facility in sighting, maintaining a low profile. A compromise between profile and automatic burst control is therefore necessary.

Most weapons are symmetrical with respect to the vertical plane, otherwise they would rotate respectively e.g. T148 semi-automatic grenade launcher. This weapon had a horizontal magazine, which caused the center of impact of rounds to shift laterally as the launcher fired successive rounds.

Applying the above formula to a typical rifle, in which "a" = 1 inch, "b" = 4 inches, "c" = 20.5 in., "M_1" = 9.5 lb., "I" = 1000 ft.lb.-sec.^{2}, "M_b" = 150 gr., "M_p" = 50 gr., "V" = 2700 fps. Firing at a full automatic rate of 750 spm, what angular deviation can be expected between rounds 1 and 2 by gunners ranging in weight from 140 to 240 lb.?

For the 140 lb. man, w = .534 rad/sec., or 534 mils/sec.

For the 240 lb. man, w = .572 rad/sec., or 572 mils/sec.

Applying the cyclic rate of 750 rpm; results in the following deviations:

For the 140 lb. man: 42.7 mils.

For the 240 lb. man: 45.7 mils.

Thus it is obvious that in full automatic fire, accuracy cannot be expected; which is normal. Also, note the small variation due to body weight; also, that the heavier body causes greater dispersion. To understand this, consider the extremes in weight; a body that approached zero weight would offer no resistance to recoil, and the only deviation would be the force about the moment arm to the c.g. At the other extreme, a rigid body, or block would cause a greater turning moment, due to the fact that the buttstock would not be free to drift rearward. (as it does against a shoulder)

Therefore, in summary, the point of contact of the buttstock with the shoulder should be in-line with, or slightly above, the bore axis. The cyclic rate should be high. Short bursts are recommended.

A rubber butt-pad is useful in reducing recoil load effects for two reasons:

(1) The rubber pad distributes the bearing load over a wider area of the shoulder,

(2) The shooter intuitively presses the buttplate harder against his shoulder, which is a favorable attitude for this purpose.

OUTLINE STUDY OF SMALL-ARMS AUTOMATIC WEAPONS

I Introduction

Scope and Purpose of Outline

This outline is a summary of all of the factors that must be considered in designing a new weapon. There may be additional factors depending upon the specific requirements of each new weapon system, but this outline represents a minimum list of factors. It may be utilized as a checklist of weapon design parameters and is provided as a guide to insure that no important weapon element is overlocked when a new concept feasibility study is conducted.

II Basic Weapon Operational Cycles (Blowback, Recoil, and Gas)

A. Simple Blowback

1. Definition of simple blowback cycle

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve.

b. Ammunition characteristics required for simple blowback cycle

(1) Case and chamber design

(a) Headspace

(b) Breeching space

(2) Case Material

(3) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages of simple blowback systems

B. Modification of Simple Blowback Cycle - Delayed Blowback

1. Definition of delayed blowback

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

II 1. b. Ammunition characteristics required for delayed blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

C. Modification of Simple Blowback Cycle - Retarded Blowback

1. Definition of retarded blowback

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

b. Ammunition characteristics required for retarded blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

D. Modification of Simple Blowback Cycle - Advanced Ignition

1. Definition of blowback with Advanced Ignition

a. Cycle of operation

(1) Construction of calculated T-D curve

(2) Kinematic and dynamic analysis of cycle

II . b. Ammunition characteristics required for advanced ignition blowback cycle

(1) P-T curve at chamber

(2) Case and chamber design

(a) Headspace

(b) Breeching space

(3) Case Material

(4) Lubrication

E. Gas Operation

1. Definition of Gas Operation Cycle

a. Cycle of operation

(1) Kinematic and dynamic analysis of cycle

(2) Construction of calculated T-D curve

(3) Gas systems

(a) Open - impingement

(b) Closed - cut off and expansion

(c) Pressure - time curves of chamber and bore at gas port

b. Ammunition characteristics required for gas operation cycle

(1) Case and chamber design

(a) Headspace

(b) Breeching space

(c) Pressure data

(2) Case material

(3) Lubrication

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

II F. Recoil Operation

1. Definition of recoil operation

   a. Cycle of operation

      (1) Kinematic and dynamic analysis of cycle

      (2) Construction of calculated T-D curves

      (3) Recoil systems

         (a) Short recoil system

            (1) "Browning" cycle

            (2) German cycle

            (3) Linked action cycle

         (b) Long recoil systems

      (4) Accelerators

         (a) Definition

         (b) Variable lever

         (c) Integral cam

   b. Ammunition characteristics required for recoil operation

      Short recoil Long recoil

      (1) P-T curves of interior ballistics system

      (2) Case and chamber design

         (a) Headspace

         (b) Breeching space

      (3) Case material

2. Strength requirements of breech mechanism

3. Advantages and disadvantages

G. Externally Powered Systems

II G. 1. Revolver and multichamber systems

a. Cycle of operation

b. Kinematic and dynamic analysis of cycles

c. Ammunition characteristics required for revolver and multichamber systems.

(1) Case and chamber design

(2) Dynamic seals

2. Multi-barrel systems

a. Cycle of operation

b. Kinematic and dynamic analysis of cycles

c. Operational cam design

3. Drive systems

a. Hydraulic

b. Electric

c. Mechanical

III Weapon Components

A. Breech Locking Systems

1. Integral locks

a. Rotary locks

b. Tilting locks

c. Sliding locks

2. Separate locks

a. Rotary

b. Tilting

c. Sliding

3. Strength requirements of locking systems

III B. Feed Systems

1. Magazines

   a. Box type

      (1) Single column

      (2) Double column

   b. Rotary types

      (1) Fixed

      (2) Removable

   c. Magazine springs

   d. Kinematic analysis of feeding operation

2. Strip and clip feed systems

   Kinematic analysis of feeding from strips and clips

3. Belt Feed System

   a. Pull out or "U" type

   b. Push through

   c. Fundamentals of link design

   d. Kinematic analysis of belt feeding systems

C. Extraction and Ejection Systems

1. Kinematic and dynamic analysis of extractor and ejection

2. Functional requirements of extractors and ejectors

D. Fire Control Mechanisms

1. Sear systems

   a. Automatic fire systems

   b. Semi-automatic fire systems

   c. Trigger pull requirements

III D. 2. Ignition systems

a. Percussion firing systems

(1) Single striker

(2) Multiple strikers

(3) Firing pin design

(a) Tip configuration

(b) Tip support at time of firing

(c) Types

Fixed - Movable - Inertia

b. Electric firing systems

(1) Electrical energy

(2) Typical circuitry

Insulation problems

(3) Firing pin design

Tip configuration

E. Barrel Design for Small Arms Automatic Weapons

1. Chamber design

Headspacing

(1) Rimmed and belted cartridges

(2) Rimless cartridge

(a) Shouldered

(b) Straight

2. Rifling

Types and twists

3. Thermal problems of barrels at high firing rates

F. Muzzle Attachments

III F. 1. Muzzle brakes

Design

2. Muzzle boosters

a. Design

b. Special considerations

3. Flash hiders and flash suppressors

G. Chargers

1. Manually operated

2. Externally powered

a. Air

b. Hydraulic

c. Electric

d. Cartridge actuated

H. Mounts - Mounting conditions with respect to weapon functioning

1. Vibrational characteristics of automatic weapons

2. Mounting conditions

a. Shoulder fired or hand held

b. Bipod and tripod mounts

c. Vehicular mounts

(1) Armor vehicles

(2) Aircraft

(a) Fixed wing

(b) Helicopter

3. Recoil adapter design

Buffer design

IV Areas of Special Consideration for Small Arms Automatic Weapons

A. Open and Closed Bolt Cycles

B. Weapon Safety Requirements

C. Economics of Fabrication

1. Sheet metal construction

2. Die castings; sintered metals

3. Special machining techniques

a. Milling

b. Broaching

c. Turning

D. Service Life Requirements

V Tactical and Logistical Requirements Affecting Small Arms Automatic Weapon Design

A. Human Engineering Factors

1. Handling characteristics

2. Operating characteristics

B. Environmental Conditions

1. Temperature extremes

2. Mud and dust problems

3. Lubrication problems

IV TYPES OF WEAPON SYSTEMS OF OPERATION

(AUTOMATIC WEAPONS)

Classes of weapon systems may be outlined, in general, as follows:

I Blowback

A. Pure blowback (Cal. .45 M3) (Cal. .22 LB)

B. Delayed blowback (Cal. .45 Reising SMG)

C. Retarded blowback (Cal. .45 Thompson SMG) (earlier versions)

D. Advanced primer ignition (Oerlikon)

II Recoil

A. Long recoil (misc. shotguns)

B. Short Recoil

1. Toggle (maxim)

2. Propped lock (Browning)

3. Tilting barrel (Cal. .45 M1911)

4. Cammed cross-bolt (M73)

5. Hinged bolt (M85)

6. Linkage (proportioned) (R. Robinson)

C. Misc. revolver types

III Gas Operation

A. Piston (rearward)

1. Impingement (M1)

2. Cut-off & expansion (M14)

B. Bleed-off (M16) (French tube)

C. Muzzle blast (Bang)

D. lever (Colt Browning)

III E. Primer actuated (AA1)

F. Piston (forward) (misc. European)

IV Misc. self-powered

A. Floating chamber (win. shotgun)

B. Set-back (impulse) (garand exp.)

C. Alternating barrel (Hughes)

V External power

A. Electric drive (Vulcan, M75)

B. Hydraulic or pneumatic (Vigilante)

C. Gear drive off vehicle (ZB-80)

Factors that determine what type of operating system a new weapon will assume are outlined following this paragraph. Note that the military requirements are the first item, and the most important.

Ammunition characteristics are physically responsible for limiting certain types of operation, due to the nature of pressure development and available power.

SELECTION OF SYSTEM OF OPERATION

1. Military requirements
2. Weapon environment
    a. Mount flexibility or rigidity
    b. Noxious gases and/or muzzle blast
    c. Permissible trunnion load
    d. Space requirements for
        (1) Installation,
        (2) Maintenance, and
        (3) Ammunition feed, spent case and link disposal

3. Ammunition

   a. Cartridge case design

   b. Internal ballistics

   c. Ignition system (percussion/electric)

4. Effectiveness

   a. Rate of fire

   b. Belt pull capacity

   c. Weight, size, and shape

   d. Reliability

   e. Maintainability

5. Cost

Blowback Operation

The most important factor in the blow-back operated weapon is the behavior of the cartridge case, since it is in motion at the instant of firing. Blowback is usually reserved for low power cartridges, and for military weapons, is most usually found in sub-machine guns firing the cal. .45 and 9mm pistol cartridges.

Cartridges for blowback weapons are essentially cylindrical; with no neck or shoulder, and very little body taper, if any. The base is sufficiently thick to support the chamber pressure for the time and travel that the cartridge case moves during blowback. It is this initial movement that limits the blowback principle to low-powered weapons with a cartridge design that can move rearward in the chamber without danger of case stretch.

Calculations for this system are quite straight-forward. The force acting rearward on the bolt equals the chamber pressure multiplied by the cross-sectional area of the mouth of the case. The bolt impulse is equal to the projectile and gas impulses.

The cartridge case is designed to resist seizure by the chamber wall at the onset of blowback motion. At this point lubrication will make a difference in function, reflecting higher rates of fire. However, the U.S. systems use unlubricated cases, while some European systems do lubricate their ammunition. The force expended in overcoming friction between the cartridge case and the chamber may reduce the energy transferred to the bolt sufficiently to cause a short recoil, or failure to feed.

As an exercise in determing recoil velocities, the following formula is usually given:

$$M_r V_r = M_p V_p + M_c 4700, \text{ where MV is impulse}$$

This impulse is given in lb. sec.

The peak velocity to which the bolt will be accelerated at the instant of bullet exit is easily found. However, a correction factor of approximately .75 should be applied to compensate for inerti effects. This is, the bolt energy going into battery must be absorbed by the blowback energy before bolt recoil begins.

The impulse may also be determined by measuring the area under the pressure time curve, and multiplying by the bore area.

After determining bolt impulse, either the bolt weight or velocity are calculated next, depending on what system requirements are known, such as: rate of fire, geometric limitations of the bolt, (hence weight), and allowable recoil travel.

The calculation of impulse is useful in determining free recoil velocities of, say, a rifle or other complete weapon at time of firing since the weapon mass will be in a status of recoil, just as the bolt mass of the blowback system.

For example, since $I = mV$ and $F = ma$, then $F = I/t = 2.3/.002 = 1150$ lb. for an average force of a 7.62mm NATO firing weapon, rigidly mounted. Of course, weapon recoiling travel and time reduces this to loads that are easily managed. For example, an average reaction of 18 msec. reduces this load to 1/9, or a little over 100#. Thus acceleration is very important in the amount of "punishment" or effective load felt by the shooter.

Advanced Primer Ignition

In this system, the bolt is moving forward at a significant velocity at the start of ignition. Therefore, considerable rearward impulse is absorbed in decelerating the bolt to a stop prior to recoil. This principle is employed in the Oerlikon family of weapons, the XM140 firing cycle, and in heavier gun tubes. The maximum efficiency of this system occurs when the closing velocity equals the opening velocity, so that the recoil velocity is then one half of a comparable plain blowback system. Since energy is a function of V^2, then reducing the velocity by 1/2 reduces the energy to 1/4; thus reducing trunnion loads by 75%.

The weapon will be lighter, firing rate can be increased, although special features must be incorporated into the cartridge case design for this type of weapon cycle of operation.

Delayed Blowback System

These systems, of which there are many variations, employ a lock to the breechblock for only a portion of the peak pressure time. After unlocking, the remaining breech pressure acts on the bolt in "blowback" fashion. Again, special ammunition design features are required, such as a heavy case head, headspace control during opening to prevent case splitting, fluting chambers, etc.

Bolt velocity is a result of several vectors. That is, the initial energy absorbed by the unlocking linkage and bolt carrier, if any. In addition, effect of residual bore pressure acts to implement the bolt recoil. This system results in higher rates of fire, particularly if the bolt weight is light.

Retarded Blowback Systems

In this system, the bolt is not positively locked, but must act against a mechanical disadvantage in opening. The mechanism employed utilizes essentially lightweight components, with a high inertia to be overcome. This is analogous to a wheel and crank slightly off a dead-center position.

A toggle joint mechanism embodies this principle. It is important that the retarding mechanism should act against a mechanism constantly decreasing mechanical disadvantage throughout the bolt stroke. In fact, the retarding element should act so that its delaying force reflects the shape of the pressure time curve; high at the start and low through most of its action. The proper location of toggle pivot joints effectively act to produce this motion. The resultant mechanism is relatively smooth in action, but must be limited to relatively low to medium powered classes of ammunition.

Since the mass effect of rotating linkages vary, the bolt motion cannot be determined in simple terms of impulse and momentum.

Set-Back (Impulse)

This is a combination blowback and recoil operation, in which the bolt is free to blow back against a locking abutment for a short stroke, in the order of .060-.090 inches. A heavier bolt carrier continues to recoil rearward through a dwell travel before it unlocks the bolt and reciprocates it. This system is limited by cartridge case design considerations.

Recoil Operation

There are many variations of this principle of operation, in which the bolt, barrel, and lock recoil together until the breech can be safely unlocked. The energy generated during this motion is then distributed throughout the cycle of operations, and the manner in which this distribution is made is a measure of the efficiency of the weapon.

The Long Recoil system will not be discussed at length because it is of limited military interest for small arms, due to practical rate of fire restrictions.

The Short Recoil mode is popular because of its universal effectiveness. In this system, the reciprocating bolt receives energy from three vectors: First, the momentum from its basic recoil motion; Second, an acceleration from the barrel, due to an accelerator mechanism that in turn helps buff the barrel stroke, and Third, an added impulse from the blowback action of residual pressure at the time of bolt unlocking. In addition, some weapons use a muzzle booster, in which the muzzle gases are trapped within an exterior fixed housing and impinge upon the barrel as though the barrel muzzle end were a piston, accelerating it further.

Some weapons, notably the Browning Machine Gun Series, fire as the bolt, barrel, and lock are ending their counter-recoil (forward) motion. The initial recoil thrust is checked, and the need for counter-recoil buffers to absorb and dissipate the forward kinetic energy of the counter-recoiling parts is eliminated. This principle is quite similar to that employed in the advanced primer ignition type of Blowback operation discussed earlier.

In the analysis of a short recoil system, a review of system forces is in order.

Consider the following schematic:

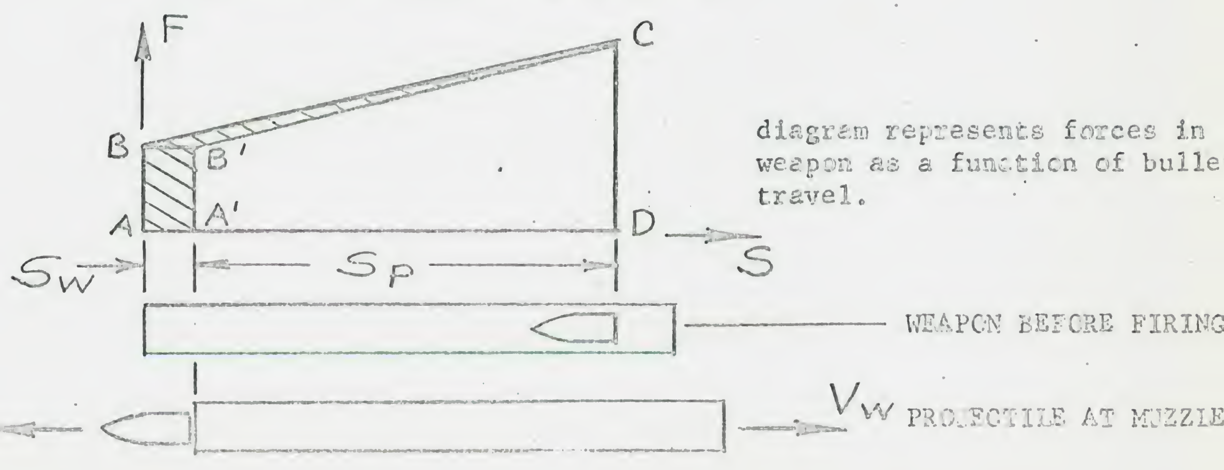


diagram represents forces in weapon as a function of bullet travel.

For the initial few milliseconds, while the masses are accelerated to their operating velocities, the following relationships are true:

Since $F = ma$

acceleration of the projectile is

$a_p = F/W_p\,g$ (W_p = projectile wgt. + 1/2 charge wgt)

while the acceleration of the weapon in the opposite direction is:

$a_w = F/W_w g$

The accelerations of the projectile and the weapon are inversely proportional to their weights:

$a_p/a_w = W_w/W_p$

The same is true of the velocities and travels of projectile and weapon:

$V_p/V_w = S_p/S_w = W_w/W_p$

Now the impulse of the projectile is at all times equal to the impulse of the weapon.

$I = mV$

$I_p = W_p/g\ V_p$

$I_w = W_w/g\ V_w$

therefore: $\dfrac{I_p}{I_w} = \dfrac{W_p \times V_p}{W_w \times V_w}$ Since $\dfrac{V_p}{V_w} = \dfrac{W_w}{W_p}$

then $\dfrac{I_p}{I_w} = \dfrac{W_p}{W_w} \times \dfrac{W_w}{W_p} = 1.$

The kinetic energy of the projectile at the muzzle corresponds to the area A'B'CD of the schematic diagram since the ordinate represents force and the abscissa is travel, and the product of force and travel represents energy.

The energy of recoil of the weapon Ew is indicated by the small area ABCB'A'. Note that the same force acts on the weapon as on the projectile but the travel (thus velocity) of the weapon is much smaller.

The energies of the projectile and weapon are in the same ratio as their travels, which is also the inverse ratio of their weights.

Therefore:

$$\frac{\sqrt{Ep}}{\sqrt{Ew}} = \frac{Sp}{Sw} = \frac{Ww}{Wp}$$

In summation, on firing, the muzzle momentum of the projectile is equal to the recoil momentum (or impulse) of the weapon, but the kinetic energies of weapon and projectile are not equal.

The sum of energy of weapon and projectile, is equal to the whole energy available. The projectile energy is expended during flight along the trajectory and at the target, while the weapon energy goes to work performing the automatic cycle of operations of the weapon.

For a short-recoil type of weapon, a general study of energy requirements is as follows:

MBBL = Mass of barrel and associated components in motion

MBO = Mass of bolt assembly

I = impulse of projectile at muzzle

Since $I = MV$ and $E = 1/2MV^2$ then $V^2 = I^2/M^2$ and $E = I^2/2M$

$E\ (BBL + BO) = I^2/2\ (MBBL + MBO)$

Using a mass ratio of $r = MBO/MBBL$

$E\ (BBL + BO) = I^2/2\ MBBL\ (1 + r)$

The energy of the Bolt above is:

$EBO = I^2/2MBBL \cdot r/(1 + r)^2$

The energy of the Barrel above is:

$E\ BBL = I^2/2MBBL \cdot 1/(1 + r)^2$

After the pressure in the barrel has decreased to a safe value (dependent upon cartridge case design) then separation of Bolt and Barrel can start.

Since the barrel assembly mass is quite heavier than the bolt, the bolt requires additional energy to perform its assigned tasks.

Here an accelerator is used which transfers much of the remaining barrel energy to the bolt. The percentage of barrel energy transferred to the bolt, in this case, "n" is assumed to be about 70%. This acceleration also serves to buff the barrel. (absorb excess barrel energy)

The maximum energy which the bolt can have is then

$$\text{Max } E_{BO} = I^2/2M_{BAR} \cdot \frac{r + n}{(1 + r)^2}$$

This energy must enable the bolt:

1. To perform work such as feeding, cocking etc. = Ew (recoil)

2. Overcome friction losses = E f (recoil)

3. Store energy in the drive spring for counter-recoil = Es

Also, the bolt will have a certain amount of energy which is partially absorbed by the buffer = E B

Thus

$$\text{Max } E_{BO} = E_w + E_f + E_s + E_B$$

For the counter-recoil stroke a similar equation is developed, so that the bolt energy at the end of C' recoil is:

$$E_{BO} (C'R) = Es + \Delta E_B - Ef (C'R) - Ew (C'R).$$

Ew (C'R) is the work done in C' recoil such as stripping the cartridge from the link, chambering, etc.

ΔE_B is the energy given back by the buffer, (coefficient of restitution) and may vary widely dependent upon the intention of the designer, and in this instance is assumed to be 50 to 60%.

However, E_B may be neglected so that the minimum energy requirement can be derived. That is, the bolt should still function without assistance of energy from the buffer.

$$\text{Max } E_{BO} \text{ (OFF Accelerator)} - E_{BO} (C'R) = E$$

Where E is all the energy used to perform the cycle of operations. Keeping this simple formula in mind, a relationship between starting bolt energy and remaining bolt energy at the end of the cycle will be developed.

Recoil time is determined by the initial velocity, which is a function of the square root of the initial energy, and the time for counter-recoil a function of the remaining counter-recoil velocity, likewise a function of the square root of the end energy on counter-recoil.

The time ratio for recoil and counter-recoil TR/TC'R = β where β is about 75 to 80%.

Therefore, since

$$TR/TC'R = \frac{\sqrt{E\ C'R}}{\sqrt{\max\ E\ BO}}$$

Then $E\ C'R = \beta^2\ (\max\ E_{BO})$

Now since $\text{Max}\ E_{BO} = E + E\ C'R$,

(That is, all energy used on recoil and c'recoil plus energy remaining at end of c'recoil.)

Putting the above two equations together,

$$\text{Max}\ E_{BO}\ (1 - B^2) = E$$

$$\text{Max}\ E_{BO} = E/\ (1 - B^2)$$

Using B = .80, then

$$\text{Max}\ E_{BO} = E/(1 - .64) = 2.8E$$

In summary, this means that the initial energy of the bolt, which is powered mainly by the barrel weight, must be about three times greater than the energy which is used for the weapon function.

The details of a mathematical analysis of a short recoil type of weapon design depends upon the mechanism design selected or invented for performing the sequence of operational functions. Therefore it is not feasible to set up an analytical method which will apply universally to all short recoil weapons.

Of course, the cartridge must be defined as to velocity, pressure-time data, and case strength.

Vallier's formula for approximating the duration of the residual pressure is

$$T_{RES} = \frac{Mc}{AP}\ (9400 - Vp)$$

P = Muzzle pressure
Vp = Proj. vel.
Mc = Charge Mass.
A = Bore area

The residual pressure time span is required in order to determine forces that act on the bolt at the moment of unlocking.

The action on the accelerator should not be too abrupt, as this will create shock loads, abrasion and wear of parts. At least 4 msec. should be given as an action time for accelerator function. Too long a time will not adequately accelerate the bolt and the firing rate will be reduced. The formula for Kinetic Energy is used to determine velocity increase in the bolt.

The change in velocity will enable forces on the accelerator to be computed by using the formula $F = m\ a$

It must be remembered that the bolt velocity must be modified to compensate for inertia effects, drag, etc. This may vary from 70 to 85% of the indicated velocity.

In counter-recoil motion, two separate approaches to the control of the barrel motion may be used, depending upon the locking system selected. In one, the barrel remains locked back until the bolt approaches it while chambering a new cartridge. Then, as the bolt is locked, it moves forward with the barrel. The Browning machine gun family is of this type.

The other approach does not latch the barrel, but allows it to return forward immediately. Here, the barrel must be damped out in its motions before the bolt is ready to lock and fire.

The K.E. of components coming into battery is usually so great as to require that the weapon fire, while the components are still biased with forward momentum. Otherwise a substantial buffer mechanism will be required. Precision in the timing of the firing mechanism strike will be required.

Gas System of Operation

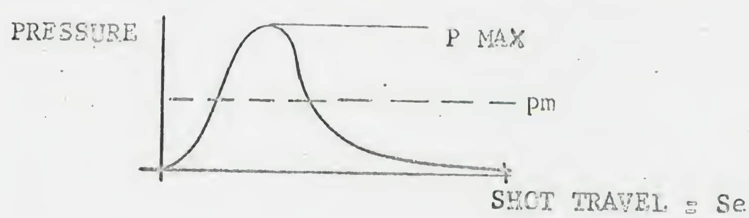
This is usually descriptive of weapons in which a transverse hole in the barrel, a gas port, is used to draw off gas, in order to operate the unlocking and other functional components.

In considering the utilization of gases vs. the effect upon muzzle velocity, there is only an insignificant loss of projectile velocity, averaging about .5%. The gas operated gun usually has an ample supply of available energy for actuating the weapon mechanism.

This is useful, particularly in the early development phase of a new weapon, if there is a lack of operating power, the gas port is merely opened up several thousandths of an inch. However, the most abused method of determining gas port size required, in the initial test weapon is to begin with an unusually small hole, approximately .030" in dia. and progressively open the hole, while measuring weapon function by a time-displacement curve of the operating rod or carrier. The many variables, from one weapon to another, of propellant characteristics (burning rate), mass ratios, moments of inertia, friction, geometry of control surfaces in sliding, render initial calculations vulnerable to errors in assumed values, so that, in the long run, the trail and error process is resorted to, within limitations.

In a gas operated weapon, the gas port should be located at least further than the "point of all burnt" of most of the propellant. If the gas port is too close to the chamber particules of unburned propellant will enter the gas cylinder and either burn, causing erratic power development, or gather in such a manner as to cause a secondary explosion during some subsequent firing.

Determination of gas pressure along the bore:



pm = mean gas pressure, where the area under pm = the area under the curve. Relating the pressure curve to energy of projectile:

$$\text{ENERGY} = pm \times A \times Se \simeq \frac{W_p + .5\,W_c}{2g} \times V^2$$

The ratio of pm to p max is called the pressure ratio "N"

$$N = Pm/P\ max$$

This is a typical empirical value of a variety of weapons and propellant types. A value of .4 is typical for an efficiently used small arms propellant. The smaller "N" is, the more efficiency realized.

The length of barrel may be expressed as a function of the pressure ratio with the formula given as follows:

$$Se = \frac{Wp + .5\ Wc}{2g\ N\ Pmax\ A} \times V^2$$

Demonstrating the 7.62mm example:

$$Se = \frac{150/7000 + 23/7000 \times 2650^2}{64.4 \times .4 \times 48{,}000 \times 144^2 \times \pi/4\ .308^2/144^2}$$

$$Se = 1.88 \text{ ft.} = 22.6''$$

<u>Characteristics of Pressure, Velocity and Time curves during Projectile Travel.</u>

A series of empirical curves have been developed from observations of the pressure curve form as adapted to the pressure ratio and projectile travel. These are all listed in the Oerlikon Pocket Book, available in most ordnance engineering libraries.

For a sample pressure ratio of .4, the following values at the moment of maximum gas pressure are:

Short travel: $S_1 = Se \times .074$

Time: $t_1 = 2\ Se/V \times .358$

Velocity: $V_1 = V \times .383$

At the muzzle:

Gas pressure: $pe = pm \times .400$

$= p\ max \times .16$

Time: $te = 2\ Se/V \times .946$

For plotting values of shot travel S, corresponding to the ratio $\lambda = s/s_e$ we can obtain the gas pressure, velocity, and time of which typical values are:

V_1; t_1; S_1 = values at point of p max

$P = p\ max \times a$

$v = V_1 \times b$

$t = t_1 \times c$

ratio λ	a	b	c
1.0	1.0	1.0	1.0
2.0	.769	1.46	1.218
5.0	.397	2.046	1.672
10.0	.214	2.453	2.267
15.0	.144	2.665	2.794
20.0	.108	2.812	3.286

With these values, as functions of λ(travel) we can caluclate the variation of gas pressure, velocity, and time with shot travel in the bore. There are variations due to changes in projectile and/or propellant weight, temperature, loading density, ignition variables, shot start, rifling dimensions, and other factors. Gas pressure is measured either by piezo-electric gauges or by copper crusher gauges. The copper crusher method is quite old, and is based merely on the permanent deformation of a copper cylinder due to a load proportional to the gas pressure.

Major caliber guns use "loose" pressure gages that are merely put into the combustion chamber, while small arms systems have "fixed" pressure gages that are mounted on a test barrel. The crusher gage is suitable for checking manufacturing of ammunition lots for comparative measurements. It is limited in that only relative results are obtained, because the evolution of gas pressure is a dynamic process, while the gauge can be calibrated only statically.

The piezo-electric method for measuring pressures is based on the property of certain crystals of developing electric charges on certain surfaces if they are mechanically stressed. Quartz is the one type most suitable for ballistic pressure measurements, because of its great compressive strength, independence of temperature, and high natural frequency. The full curve of gas pressure in the combustion chamber can be reflected by piezo-electric equipment.

Measurements may be determined at the selected gas port location (in a test barrel) in order to determine variations, if any, between lots of ammunition. However, the pressure data after the gases throttle turbulently through the gas port and expand in the gas cylinder are only of academic interest. They should be of utmost important to the designer, but usually only the pressure at the gas port is measured.

In selecting a gas port location, power requirements are of primary concern. Gas temperature should not be too high, as this will cause gas port erosion and resultant erratic power development, and it should not be too low, with a port location far down the barrel, or the gas cylinder will foul too quickly, particularly in a cut-off and expansion gas system. Location of the gas port also effects the time delay between ignition and start of operating rod travel, in turn, affecting dwell travel before the bolt begins to unlock.

The following variables affect operating characteristics of a gas operated weapon:

1. Location of gas port

2. Piston diameter

3. Initial volume of the gas cylinder

4. Weight of primary components (operating rod assembly)

5. Weight of secondary components (bolt assembly)

6. Dwell travel between primary and secondary components.

7. Ammunition characteristics, including cartridge case design as well as interior ballistic data.

8. Finally, the gas port dia., which can function as a control element, but only within limitations.

Of the piston operated types of gas systems there are two basic types, the "impingement" system and the "cut off and expansion", or White, system. In the former, gas acts directly on the piston end of the operating rod after passing through the gas port, while in the latter, the gas cylinder incorporates a large volume that the gas must fill before the hollow piston begins its motion. After the piston has moved a short distance, the gas port in the barrel is blocked by the piston wall, so that the previously trapped volume of gas in the piston cavity does the work of driving the piston by its expansion. The first type was used on the rifle, M1, while the second is used in the rifle, M14 and machine gun M60. The latter type has the advantage or providing a limited measure of compensating for differences of pressure between different lots of ammunition. That is, a higher bore pressure would cause the piston to move back quicker, causing the gas port to close earlier, limiting the amount of gas that entered the gas cylinder.

Some military rifles have been developed that incorporate a method of manually adjusting either the gas port or some element of the gas system. This is intended to adjust the gas power for differences in environment, wear, and/or ammunition. Devices of

this type should be avoided if possible, as field experience indicates that it will only be accidental if the weapon is adjusted properly. That is, after a G.I. has increased power to adjust for sand, dust, fouling, etc., he is not likely to decrease power to the proper level after cleaning his rifle. The result will be excessive velocity of moving parts, with attendant battering, wear, and possible breakages.

The components accelerated directly by the gas are referred to as the primary mass, and includes the piston, operating rod, and any other directly connected mass, depending upon the weapon design. As this primary mass moves to the rear, it unlocks the bolt and carries the bolt assembly, which is called the secondary mass. For smooth functioning, the primary mass should be considerably heavier than the secondary mass. That is, the recoiling operating rod or bolt carrier develops a given kinetic energy from the gas pressure. When it unlocks the bolt and starts carrying it rearward, kinetic energy is transferred from the operating rod to the bolt. This is reflected in a sudden drop in operating rod velocity. The nature of this velocity shift determines the degree of inertia loss of the primary mass and if this is high, function will be erratic. The primary mass divided by the secondary mass is called the MASS RATIO, and this should be at least 3. As a note, the Soviet AK-47 weapon has a mass ratio of 5.4, a highly impressive figure.

Residual pressure in the chamber should act to help the secondary mass, in the work of accelerating the bolt to its peak recoil velocity. For this reason, mere gas port adjustment does not efficiently constitute proper gun development, as the timing of the moment of bolt unlocking can improve the efficiency of the weapon cycle.

In the continuation of rearward motion, the operations of extraction, ejection, cocking, and feeding are performed. At buffer contact, 40 to 60% of the remaining energy is returned by the buffer to assist the drive spring in maintaining the required rate of fire. The closing, or counter-recoil slide velocity provides energy to strip the cartridge from the link, chamber the cartridge and lock and fire the weapon. At slide closing, the velocity should not exceed 8 fps or a condition of slide rebound may prove to be a problem.

In evaluating a gas system, consider the fact that the operating piston load of necessity is eccentric to the bore. Therefore frictional force between the operating rod, bolt, and receiver, or frame, may be high unless adequately long bearing surfaces are provided. The frictional drag is aggravated by marginal lubrication conditions due to gas fouling on the piston and cylinder after extensive firing.

Attempts have been made to incorporate annular pistons concentric to the barrel, but barrel heating limits the effectiveness of this approach in extended firing.

The forces in the recoiling slide bearing on the receiver can be solved thus a series of moment equations.

The standard three equations of equilibrium $\sum F_V = 0$; $\sum F_H = 0$

$\sum$ Moments $= 0$ should balance.

As a matter of detail, in the gas piston design there must be a minimum clearance of several thousandths between piston and gas cylinder to permit function with thermal expansion. This will allow a small amount of gas leakage, which can be minimized by incorporating grooves on the piston. These grooves help block gas escape by the fact that turbulant flow in the grooves acts to keep the piston concentric in the cylinder, and the turbulence blocks the flow of gas to a marked degree.

Note that with the formulae K.E. $= 1/2m\ V^2$ and $I = mV$ then K.E. $= I^2/2M$

Therefore, it is seen that K.E. is inversely proportional to the mass. As a result, the lighter the piston and operating rod the higher the energy. This is due to the effect of V^2, but the operating rod weight must be balanced against bolt weight, the required stroke with the rate of fire, and momentum with work. That is; too light an operating rod will create frictional forces due to the high velocity, that will dissipate operating rod energy quicker than a slower-moving, heavier operating rod.

The required velocities of moving parts to attain the desired rate of fire, of course, is a function of all the variables of gas port location and size, piston diameter and stroke, and mass and stroke of operating slide and bolt. The common peak velocities used are approximately 32 feet per second, but may range from 24 fps up to 50 fps, depending upon design requirements. The location of slide guide ways with respect to center of gravity, piston and bore is quite critical with respect to loads on the receiver, location of return spring and buffer. Moment arms should be minimized wherever possible.

As for the operating rod energy, after estimates have been made of the required peak velocity, this can be related directly to the piston size. Using $I = m\ V$, the required impulse is determined. Integrating under the pressure time curve between the time the bullet passes the gas port and the time that pressure is cut-off (or exhausted at the muzzle) determines available power.

Gas piston area is $\left(\text{Impulse required}/\text{Impulse available}\right) \times 1.$ in. sq.

P = Pressure
A = Area
M = Mass
dt = time diff.

Also, change in velocity of the piston is a function of $\frac{PA}{M}$ dt

Then, after dwell travel, the operating rod momentum will unlock and carry the bolt back. At this instant their combined momentum will be slightly lower than the single operating rod momentum because of the work done.

Typical Analysis of Gas Systems

A brief resume of factors affecting gas system design was given in the previous chapter of this series; however, a number of particular studies of gas systems analysis will be reviewed here, in order to provide a comparative basis, or guideline, for other gas system studies.

The studies reviewed are as follows:

(A) Energy extracted from the bore to operate a 15.mm Spotting Rifle

(B) Effect on weapon function of Ammunition Pressure Variance (7.62mm)

(C) Maximum Pressure in M14 gas cylinder

(D) Maximum Pressure in M60 gas cylinder

NOTE: These studies are based upon actual instumented values, and therefore are realistic.

15mm Spotting Rifle, XM90 Gas System Analysis

The spotting round was developed so that through proper selection of ballistic coefficient (projectile weight, caliber, form factor) and muzzle velocity the trajectories of spotter and major weapon would match at long ranges. Much of the preliminary ammunition development was conducted using single shot Mann barrel test fixtures, so the object of this study is to determine how much variation in velocity is caused by tapping some of the gas from the bore and into the gas cylinder.

Since the projectile is affected by gas taken from the bore before projectile exit, the pressure-time history of the gas cylinder up to the time of projectile exit will indicate the magnitude of the energy extracted from the interior ballistic system of the weapon.

Calculated Energy extracted:

Impulse = Ft = APt

$I = 1.59$ lb. - sec.

$E = I^2/2M = \dfrac{I^2 g}{2W}$

$E = \underline{5.83 \text{ ft.-lb.}}$

A = Gas piston area

$= \pi/4\ (.765)^2 = .459$ in.^{2}

P = Gas cylinder Pressure

= 2800 PSI

t = 1.2 msec.

W = 7.0 lb. (Inertia)

Piston velocity at time of bullet exit = 7.2 fps. This continues to accelerate to a maximum of 11.6 ft./sec. after bullet exit)

$$E = \frac{m V^2}{2} = \frac{W V^2}{2g} = \frac{7. \times 7.2^2}{64.4} = \underline{5.64} \text{ ft.-lb.}$$

The gas system functions at an efficiency of approximately 30%, therefore/comparing energy delivered to the inertia mechanism vs. energy extracted from the bore, we have

$$E_c = \frac{5.83}{.3} = 19.5 \text{ ft.-lb.}$$

Muzzle Energy of Projectile:

$$E_m = \frac{Wp\ Vm^2}{64.4}$$

Wp = 1240 grains

Vm = 1750 fps

$$= \frac{1240 \times 1750^2}{7000 \times 64.4}$$

$Em = \underline{8423 \text{ ft.}}$ - lb.

Percentage Energy loss

$$Ls = Ec/Em \times 100 = \frac{19.5}{8423} \times 100 = .232\%$$

This is verified by data taken from six different lots of ammunition, in which five lots showed no significant difference in velocity loss, measuring with and without gas system. In fact, the velocity variation for the test barrel without the gas system was greater than the variation going from "no-gas-system" to "gas system".

Therefore, other factors, such as powder measure, rifling tolerance, and projectile tolerances offer greater deviations in muzzle velocity than the energy given to the gas system.

Effect on Weapon Function (M14 Rifle) of Ammunition Gas Pressure Variance at the Gas Port

This is a brief study of the effect on weapon function caused by significant variations in gas pressure at the gas port area.

This study should result in a determination as to what pressure variations at the gas port positions are acceptable for weapon function.

It will be shown that a relationship exists between the pressure in the barrel and the pressure in the gas cylinder together with the velocity of the operating rod.

In the function of the gas cut-off and expansion system, as utilized in the M14 rifle system, there are two different periods. The first is the movement of the piston and the operating rod, until the cut-off, and the second is the ploytropic expansion of the gas in the gas cylinder after cutoff. The study deals only with the first period, which will give an adequate relationship between the gas pressure in the cylinder and the velocity of the operating rod with respect to the pressure in the barrel at the time of cut-off.

The velocity of the gas flow "W" through the gas port is constant because the pressure in the cylinder is under the critical pressure.

Therefore, equation #1 is:

(1) $W = \sqrt{2g \, Cp \, T_1 \, \Delta}$

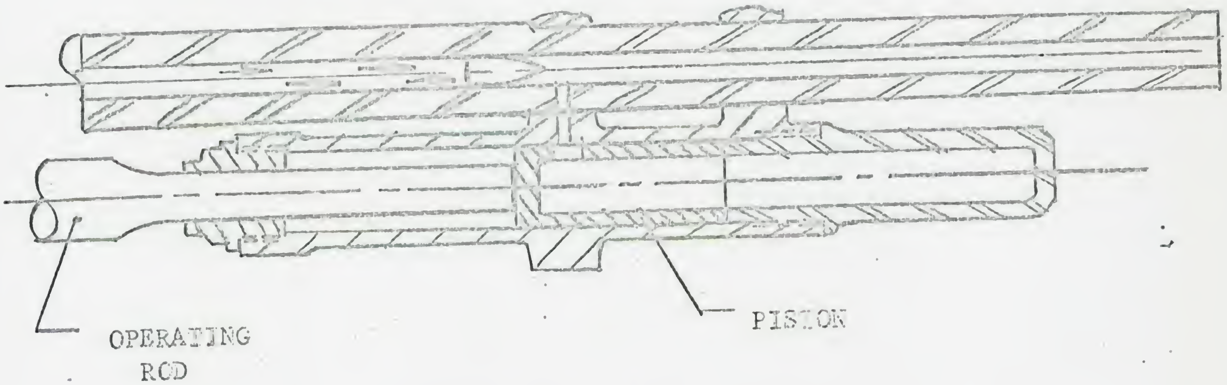
(1a) $\Delta = (P_2/P_1)^{2/K} - (P_2/P_1)^{(K+1)/K}$

(1b) $P_2/P_1 = (f) \left[2/K+1 \right]^{K/K-1}$

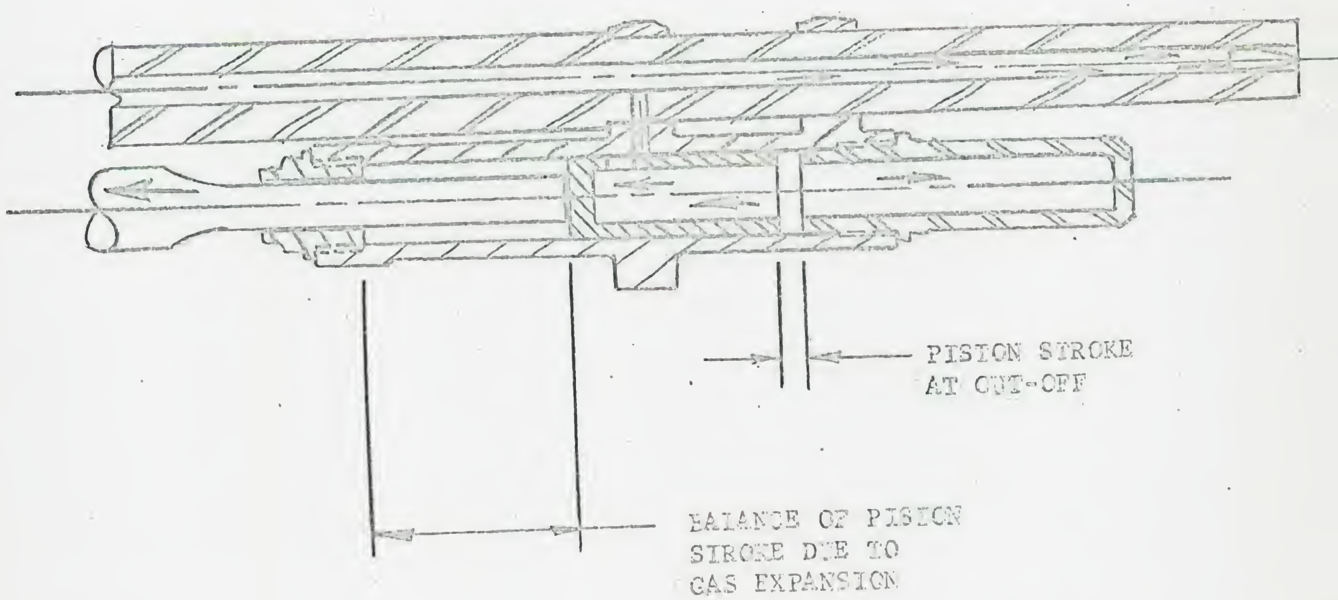
W = Velocity of the gas
Cp = Specific heat of the gas
$P1$ = Pressure in the barrel
P_2 = Pressure in the gas cylinder
K = exponent of polytropic expansion
T_1 = gas temperature in the barrel

CUT-OFF & EXPANSION

GAS SYSTEM OF OPERATION



BULLET NEAR GAS PORT



It should be noted that P1 is not the instantaneous pressure when the bullet passes over the gas port, but is the average pressure between bullet passage and time to cut-off. This will not induce an error, since it is assumed that P1 is proportional to the bullet passage pressure. This study will deal with qualitative proportions, rather than absolute values.

To determine realvalues, the equations require coefficients that are found only by experimental tests. These values were not available at the time of this study, and will be noted as the formulae are developed.

The mass flow of gas per unit time is given as:

(2) $\overset{\circ}{M} = \frac{Ao}{V2} W \qquad V2 = CG - Volume$

and can be written thermodynamically as:

(3) $\overset{\circ}{M} = \frac{Ao\ P1}{\sqrt{RT1}} \times \emptyset\ (K)$

(3a) where

$$\emptyset = \sqrt{\frac{2g\ K}{K-1}\ \Delta}$$

Ao = area of gas port which is fully open until cut-off time.

After a period of time "t" the gas weight "G" in the cylinder is:

(4) $G = \frac{Ao\ P1}{\sqrt{RT}} \cdot \emptyset\ (K)\ t \qquad$ note that $G = \overset{\circ}{M}\ t$

The pressure P2 in the gas cylinder is:

(5) $P_2 = \frac{Ao\ P1\ R}{\sqrt{RxT_1}Vo} \cdot \emptyset\ (K)\ T_2 \cdot t$

Vo = Volume of the piston cavity
(Note relationship of Vo with P2)
T2 = Temperature of gas flowing through the orifice

Another thermal relationship is

(5a) $T_2/T_1 = 2/K+1$

which produces the final expression:

(5b) $$P_2 = \frac{Ao}{Vo} \cdot P_1 \sqrt{RT_1} \cdot \psi(K) \cdot t$$

(5c) where $\psi(K) = 2/K+1 \cdot \emptyset(K)$

The maximum pressure in the gas cylinder, therefore, is at the time of the cut-off.

This time can be determined by the equations of motion for the piston and operating rod.

(6) $$M\ddot{x} + F_o + Kx = \dot{P}_2 tA$$

M = mass of piston and operating rod
Fo = Spring force
K = Spring rate
A = Gas cylinder area

$K\bar{x}$ can be neglected since the displacement $\bar{x}$ is quite small.

The solution of (6) is given by:

(7) $$X = \frac{\dot{P}_2}{6M} \cdot t^3 - \frac{Fo\ t^2}{2M}$$

Velocity is given by:
$V = dx/dt$

(8) $$V = \frac{P_2\ t^2}{2M} - \frac{Fot}{M}$$

The equation is limited, of course, by the diameter of the gas port in the piston Dp. Therefore, $X_{max} = Dp$

(7) $$\frac{P_2\ t^3}{6M} - \frac{Fo\ t^2}{2M}\ DP$$

For all practical purposes Fo (Spring force) can be neglected.

The equation then reduces to:

(9) $$tc = \sqrt[3]{\frac{6\ Dp\ M}{P_2}}$$

As a result of inspecting equations (5), (7), (8), and (9) the relationship with bore pressure P1 is as follows:

(10) $\max.\ P2 = \left(P_1\right)^{2/3} \times$ CONSTANT COEF.

(11) $V = \left(P_1\right)^{1/3} \times$ CONSTANT COEF.

The constant coefficients are a function of temperature, effective area of orifice, orifice length, propellant, and mass.

V is the operating rod velocity at cut-off.

Therefore, from equations (10) and (11) it is seen that for a pressure variation of P1 of 30%, the variation of gas cylinder pressure Po will be 20% and velocity variation V will be 10%.

Maximum Gas Cylinder Pressure (M14 Rifle)

Applying numerical values as follows:

$T = 2800\ ^{o}K$
$R = 119$ ft./degree Kelvin
$K = 1.246$

Gas port dia. = .073 in.
Dia. cylinder = .50 in.
Mass of piston and rod = .018 lb. - sec^2/ft.
Initial cylinder volume .291 in^3
The barrel pressure at the port, P1, is in the order of 15,000 psi, but values will be shown for pressures of 10,000 and 20,000, as well as 15,000 psi for comparative values.

P (bore) = 10,000 psi	P (cyl) = 1540 psi	V = 13.5 fps
P (bore) = 15,000 psi	P (cyl) = 2170 psi	V = 15.0 fps
P (bore) = 20,000 psi	P (cyl) = 2750 psi	V = 17.2 fps

Maximum Pressure in M60 Gas Cylinder

For comparative purposes, the same method may be applied to the M60 machine gun, since that weapon also employs the expansion and cut-off system.

	M60	M14
Diameter of gasport	.135 in.	.073 in.
Cylinder dia.	.875 in.	.50 in
Initial cylinder volume	1.879 $in.^3$	.291 $in.^3$
Weight inertia	1.53 lb.	.59 lb.
Piston motion to cut-off	.19 in.	.075 in.

The pressure in the barrel for each weapon is approximately 15,000 psi.

from equation (5a):

$$P2 = \frac{Ao}{Vo} \cdot \text{CONSTANT COEF.}$$

Ao = area of gas port
Vo = initial cylinder volume
P_2 = pressure in gas cylinder

Relating M60 values to M14 values ;

$$(60)\ P_2 = \frac{(60)\ Ao}{(14)\ Ao} \cdot \frac{(14)\ Vo}{(60)\ Vo} \cdot (14)\ P_2$$

(14) = M14
(60) = M60

therefore,

$$(60)\ P_2 = 3.42 \cdot 1/6.5 \cdot (14)\ P_2$$

Since (14) P_2 = $\underline{2170 \text{ psi}}$, then (60) P_2 = $\underline{1150 \text{ psi}}$

M60 gas pressure is then adjusted for the loss at the forward vent hole:

$$P = P_2\ (1 - .055) = \underline{1090 \text{ psi}}$$

Max force on piston:

$$F = \pi/4\ D^2 \cdot P = \underline{660 \text{ lb.}}$$

Definition of terms:

W = Velocity of gas flow
g = gravity, = 32.2
Cp = specific heat of gas
T_1 = Temperature gas in bore
Δ = Pressure ratio coefficient
K = Exponent of polytropic expansion
P_1 = Bore pressure (average at port)
P_2 = Gas cylinder pressure
$\dot{M}$ = Mass of gas
Ao = Area of gas port
V_2 = Covolume of gas
R = Gas constant PV/T
$\emptyset$ = function of g, K, and Δ
G = Weight of gas
t = time
Vo = Volume of piston cavity
T_2 = Temperature of gas through orifice
γ = Coefficient of K
M = Mass of inertia parts
A = Area of piston (gas cylinder)
tg = Time to gas cut-off
Dp = Diameter of gas port

V Weapon Design

In this section a number of ordnance design practices are reviewed which should serve to instruct the uninitiated. In this respect a number of elements of "art" are practiced, but ordnance engineering is predominantly a "science" as are most branches of mechanical engineering.

Experience, therefore, will lead the ordnance engineer toward utilizing the principles disclosed in these chapters. Creativity, or inventiveness, cannot be taught, but comes with practice in solving problems and "thinking out" new approaches to problems that are either old or new. This is therefore an individual quality which each engineer must cultivate.

Headspace

"Headspace" is simply the critical dimension that relates the chamber size to the cartridge size. In order that prolonged bursts of automatic fire may be executed without stretching or splitting the cartridge case the headspace dimension in the weapon must be maintained and not be affected by temperature, strain, or fatigue.

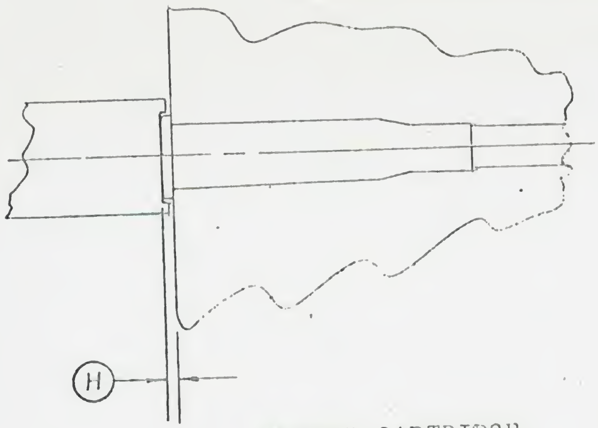
The term "headspace" was derived as a specified distance, or space, between the bolt face and barrel face, in the days of rimmed cartridges. Early cartridges, including those of high power class, were rimmed; therefore headspace indentification and control was simplified.

The use of rimless and semi-rimmed cartrdige cases necessitated other techniques for identifying "headspace" in a weapon. It, therefore is designated as the distance from the bolt face to the surface that stops forward movement of the cartridge case in chambering.

The standard necked case will have a reference diameter near the middle of the tapered shoulder area that corresponds with the same diameter in the barrel. The variations between projectile headspace and weapon headspace is the clearance, and this is usually the effect of tolerance accumulation. A number of machine guns have been developed that utilize an adjustment nut to draw the barrel closer to or further from, the breechblock face, thus compensating for changes in headspace that may be due to wear, replacement of components, case stretch indications, etc. Modern machine methods permit production of weapons with a "fixed" headspace, such as the M60 machine gun.

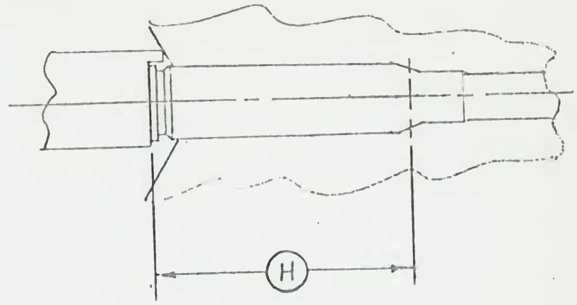
As mentioned above, the headspace in modern chambers is the distance from the locked breechface to a reference plane along the cartridge shoulder. To measure, or check, this a special set of

BASIC FORMS OF HEADSPACING



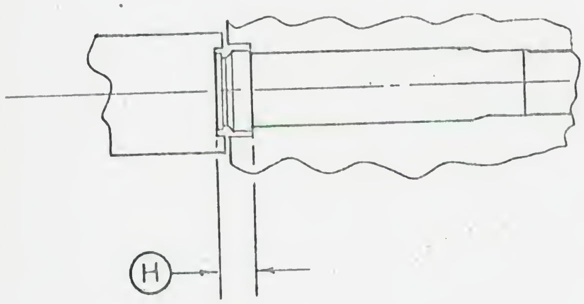
RIMMED CARTRIDGE

(30-30 Win.)



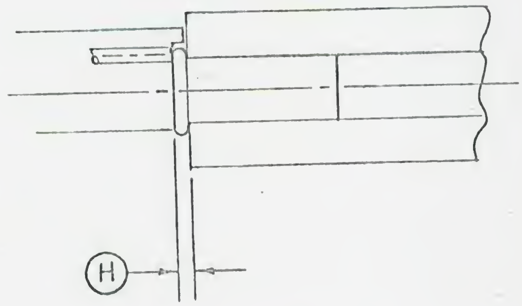
RIMLESS CARTRIDGE

(7.62 mm NATO)



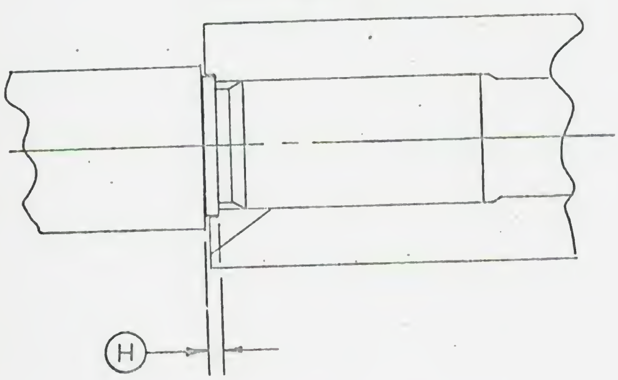
BELTED CARTRIDGE

(.375 H&H Magnum)



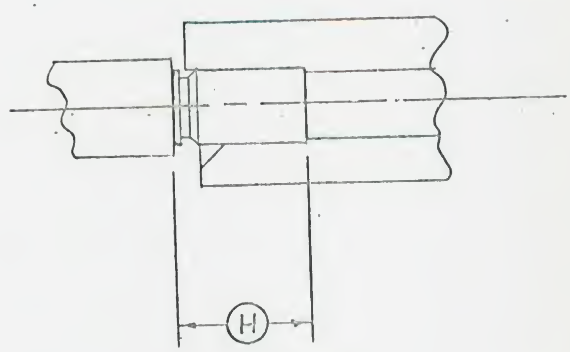
RIMFIRE CARTRIDGE

(cal..22)



SEMI-RIMMED CARTRIDGE

(cal..38 ACP)



RIMLESS CARTRIDGE

(no shoulder) (cal..45 ACP)

(H) indicates HEADSPACE CONTROL DIMENSION

headspace gages is required. A "master ring" is used to check headspace gages..

For example, in the cal. .30 M1, the headspace specified is 1.940 MIN, 1.944 MAX., a variation of only .004. This is measured from the breech face to a reference diameter of .4425 on the shoulder taper of 34°26' incl. An additional .002 headspace is permitted when inspecting overhauled rifles, with yet another .004 additional permitted as a field headspace limit for serviceable rifles, of 1.950. This is a total of .010 headspace variation permitted. This is close to the limit for a brass case, in which .016 stretch would adversely stretch the case body at the area behind the shoulder.

Another headspace control technique utilizes a "belt" or shoulder that functions as a secondary rim in front of the base rim. This short stopping shoulder stops on a ledge in the chamber shortening the headspace length, thus minimizing deflections in firing.

Control of headspace is necessary, because excessive headspace will cause irregular ignition as well as overstretched cases. Excessive headspace may cause case rupture, resulting in a stoppage, if not serious breakage. Excessive plastic deformation of the cartridge case may be observed as a shiny circle approximately 1/4"-3/8" behind the cartridge shoulder inside the case.

The following stress-strain diagrams demonstrate the conditions of either clearance or case interference, after firing, in the radial direction:

The following assumptions are made:

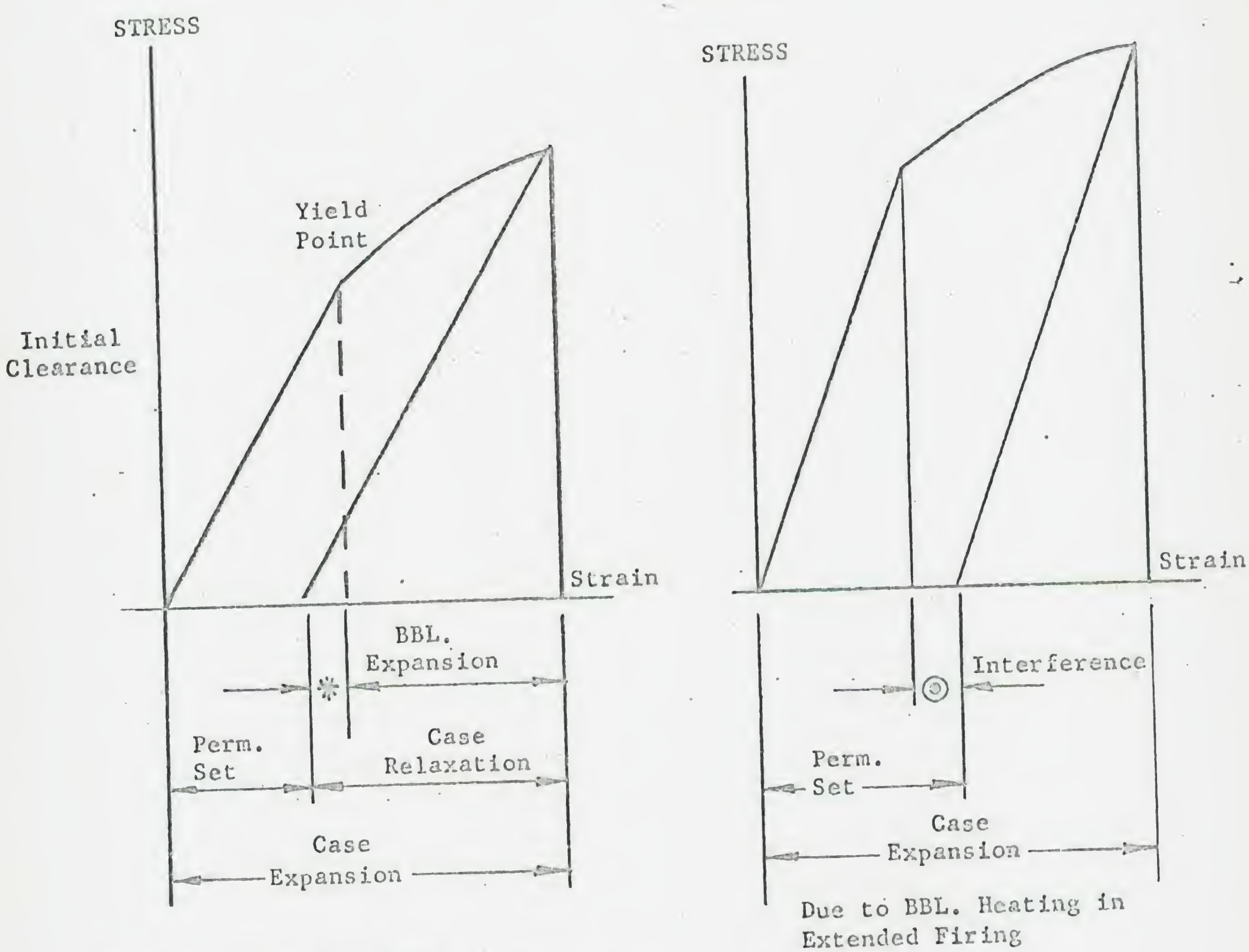
(1) The chamber recovers completely from the deformation due to propellant gas pressure, otherwise the chamber would be most improperly designed.

(2) The yield strength of quarter-hard 70-30 cartridge case brass is 40,000 psi.

RADIAL EXPANSION

CARTRIDGE CASE VS. CHAMBER

Stress-Strain diagrams of cartridge case radial expansion in chamber during firing for conditions of normal and excessive barrel expansion.



* = Resulting Clearance
(Case will be free for extraction)

The obvious method of preventing interference is to increase the chamber wall thickness or reduce case/chamber clearance.

Longitudinal clearance, of course, can cause serious stoppage, if excessive.

In a number of systems, crunch-up is intentionally designed into the case and chamber. Sufficient residual energy must be available in the moving breechblock to perform this work. (M75) weapon, as an example exhibits this method of cartridge control in the chamber. One approach to chamber design is to assign a metal-to-metal contact between a minimum cartridge and a maximum chamber. If there is not available energy to chamber a round under conditions of interference; then the chamber must be deepened, at the cost of excessive headspace when tolerances result in an excessive clearance.

Chamber sidewall taper is calculated so as to assist in ready extraction. A slight primary extraction effort will free the case from the chamber wall.

A forcing cone at the start of the rifling prepares the rotating band of the bullet for engraving by the rifling, with a minimum of free run, or "bullet jump" to minimize impact.

The Browning Machine Gun, for example, required headspace adjustment each time the weapon was reassembled or each time another barrel was inserted. The M60 machine gun has "fixed headspace" and does not require field adjustments. This is due to coordination between gun design and production techniques.

For example, in the BMG too tight headspace would cause:

(1) Failure of lock to enter recess in bolt.

(2) Failure to fire, the bolt not reaching the firing position

(3) Failure to reach and extract cartridge from the belt.

(4) Sluggish action due to excessive locking friction. This is the most frequent consequence of tight headspace.

Loose headspace would tend to cause:

(1) Excessive battering of the lock, locking recess, and lock cam.

(2) Ruptured or separated cases.

(3) Poor shot patterns due to pressure leakage at the breech.

American cartridge brass is softer than the European brass, to allow greater headspace variation. U.S. ordnance practice is for 100% interchangeability of components, whereas European practice is for greater use of selective assembly and individual fitting at assembly. For this reason, the European hard brass is not usually given to excessive case stretch.

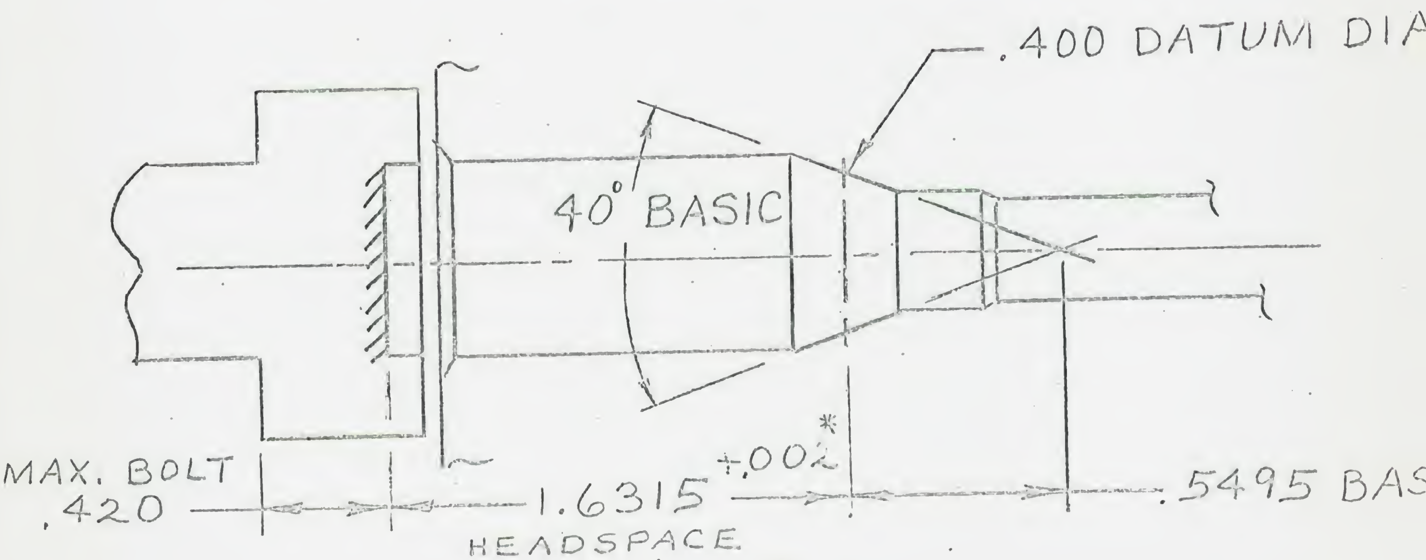
Front locking lugs are desirable for designs that tend to reduce excessive case stretch in firing. Rear locking lugs, or locking surfaces located a long distance from the case seat cause stretching of the receiver or frame and compression of the bolt body, both according to the formula $\delta = P\ell/AE$. Headspace change also occurs through temperature change over this length. For this reason, most modern weapons are front-locking rotary bolts with fixed headspace, resulting in less distortion of the cartridge case in extended firing.

Symptoms of excessive headspace are obvious in visual inspection. A case stretch is indicated by a bright zone extending all around the case about a half inch from the base. As headspace increases a fine crack will begin to show up on the cartridge body.

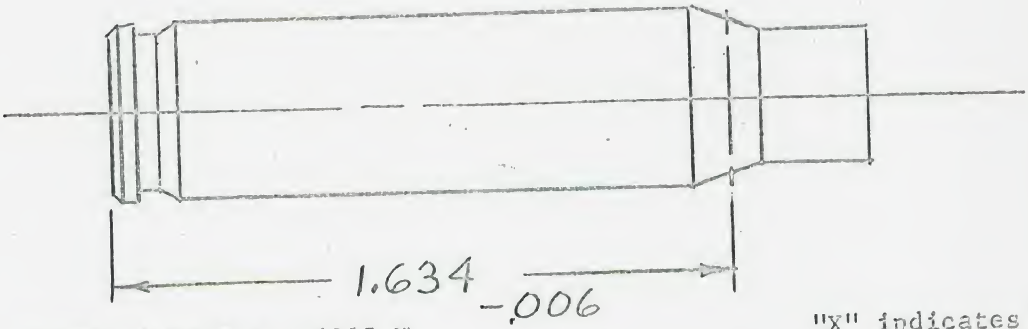
Another danger of excess headspace is in the free run it permits the cartridge case and bolt in setting back against the locking abutment. This impact intensifies the shock loading factor, and is dangerous for receivers tempered to a high hardness. The rimless cartridge can characteristically be driven forward an excess amount by the bolt slamming into battery, particularly if there is excessive forward clearance between the bolt and the barrel face.

Here a practical exercise is given in studying dimensions and accumulating tolerances in a typical weapon.

As an example, study M60 headspace diagram as follows:



Compare this with cartridge case dimensions of:



chamber 1.6345 X 1.6315 N
case 1.628 N 1.634 X

.0065 X .0025 INTERFERENCE, OR CRUSH.
CLEARANCE

"X" indicates maximum di
"N" indicates minimum di

* 1.6315 + .003 AFTER PROOF FIRING

Looking further into the development of this headspace data, study the ability of the case to stretch: The section just to the rear of the case shoulder tapers from a thickness of .010 inches to .030 in a length of 1.310 inches. The formula for determining allowable stretch is:

$d = \frac{sl}{E}$ where s = yield stress of 40,000 psi

E = 18,000,000 psi

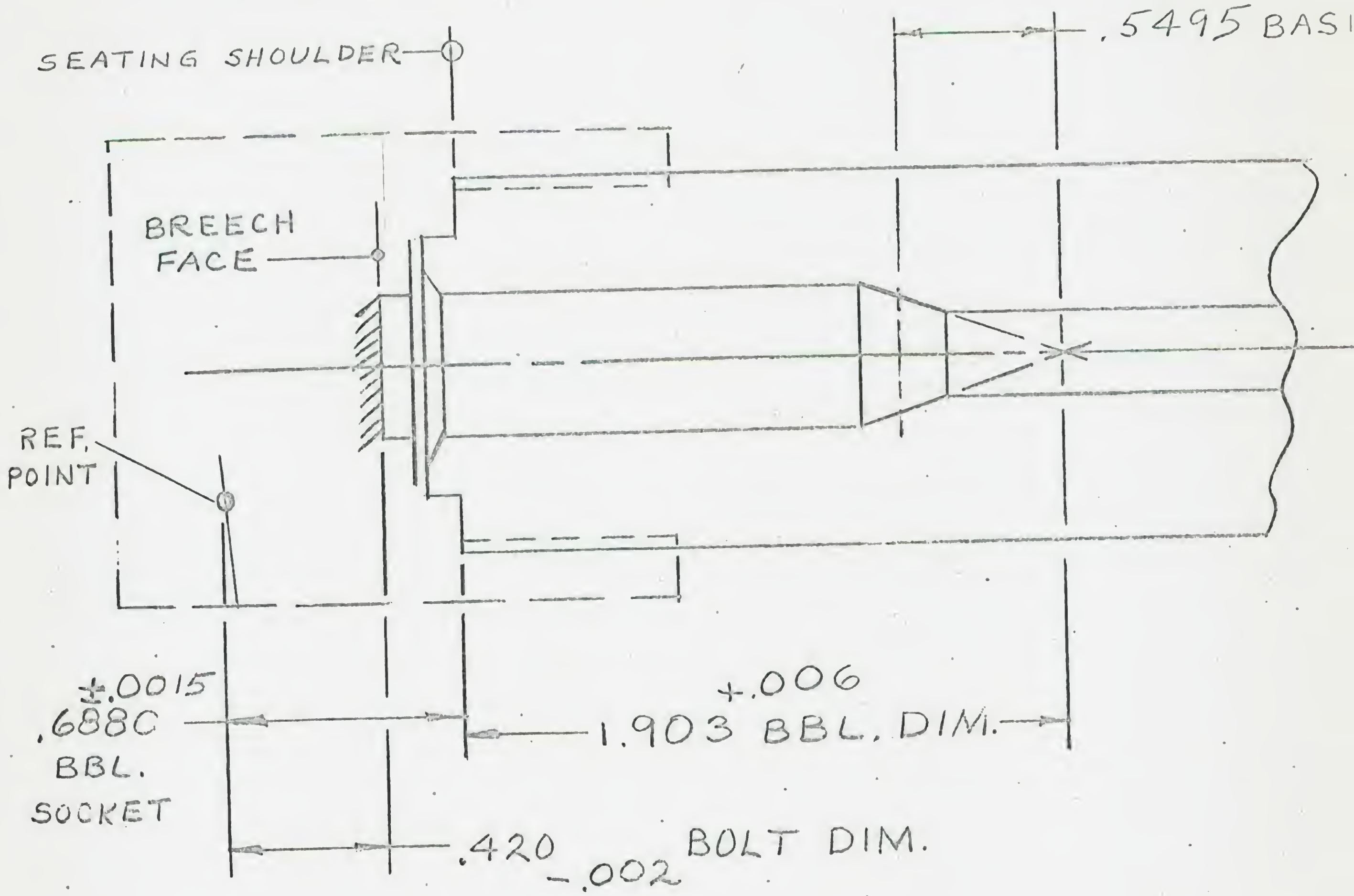
l = 1.31 in.

$$d = \frac{.404 \times 1.310}{18.}$$

$d = .003$ "

Therefore, any stretching over .003" permanently deforms the case, which illustrates why handloaders resize their cartridge cases.

A look at the individual dimensions in the M60 Machine Gun to demonstrate how tolerances can accumulate:



ACCUMULATION OF TOLERANCES:

1.909X	1.903 N	.420 X	.418 N
.6895X	.6865N	.5495	.5495
2.5985X	2.5895N	.9695X	.9675N
- .9675N	.9695X		
1.6310X	1.6200N		

Therefore, after the closest tolerances, reasonably maintained in production, the final headpsace dimension, reamed at assembly, compares as follows:

1.6335X	1.6315N	
1.6200N	1.6310	
.0135X	.0005N	METAL REMOVED

For machine guns, the barrel chamber area may increase in temperature by as much as 850° F.

The temperature expansion, then is:

$$\delta_T = E(\Delta T)\ell \qquad d = (.0000065)(850)(1.63)$$

$$d = .009''$$

It is obvious that for prolonged firing excessive stretching will ensue, which may be compensated for in the cartridge case physical properties, which are generally as follows:

Properties of cartridge brass:

70% Cu

30% ZN

	HARD	SOFT
Tensile Strength kpsi	76	47
% elongation in 2 in.	8	62
Yield strength (5% extension under load) kpsi	63	15
Rockwell Hardness	B82	F64
Melting Point	1750°F.	
Density $\#/ft^3$	532	
Coefficient of expansion per $°F \times 10^{-6}$		11.1
Thermal conductivity ($BTU/hr/ft^2/ft/F°$)		70

U.S. cartridge brass is approximately quarter hard in order to take advantage of the high percentage of permissible elongation (ductility)

In closing this topic it may be recalled that during days of adjustable headspace machine guns, a gunnery instructor could adjust the weapon so that, after firing briefly, cases would begin to split, and the students would have to quickly correct headspace, in the dark, in order to complete the course.

Typical Factors of Safety in Small Arms Design

The determination of allowable stresses for small arms components is quite important in that small arms design demands a minimum of weight and volume commensurate with the work being done by the weapon mechanism. The factor of safety is generally an educated guess and is based upon the endurance life of the subject weapon. It should take into account the non-uniformity of the material and its heat treatment, inside corners of various degrees as stress raisers, tool marks, grain direction, and a myriad of other factors.

If the allowable stresses selected in design are too high, the service life of the component is compromised, whereas if it is too low, a penalty is paid in excess weight, particularly for an infantry weapon.

One comparative design area that may be used for study purposes is the bolt locking lug design. This area is usually highly stressed, since the bolt lug size is preferably kept to a minimum. This is because, for rotary and tilting bolts, if the rotation to lock/unlock is minimized, the cam profile will be reduced, and bolt carrier bulk minimized. The locking lug height is minimized, as this determines the size of the receiver or locking ring. As a result, allowable stresses are as high as any other component in a weapon; with the possible exception of the barrel, which has additional stresses due to temperature, vibration, external loads, etc. Therefore, locking lug design practice is more representative of a small arms design structure than barrel design.

Accordingly, maximum stresses and resultant factors of safety for three bolt lug designs are compared.

These include:

(a) 7.62mm M60 Machine Gun
(b) 7.62mm M14 Rifle
(c) Cal. .30 M1 Carbine

The maximum firing force is assumed to be the maximum service pressure, multiplied by the maximum inside area of the cartridge case, since the wall of the case does not transmit bearing forces. (Except in the instance in which the case ruptures from excessive headspace).

Stresses calculated are typically shear, pressure, and bending. The shear and bending stresses are then combined into a comparable multi-axial stress, according to the Hencke-Von Mises theory.

According to this theory, when there is a two axial stress consisting of a normal stress and a tangential (shear) stress the equivalent stress is:

$$Se^2 = S^2 + 3\tau^2$$

S = BEARING
τ = SHEAR

The equivalent stress must be smaller than the allowable stress in tension. Therefore, the maximum shear stress is equal to .57 of the allowable stress in tension. (as a maximum)

It is recommended that the actual shear stress is 1.5 times the calculated average shear stress, because in bending, the shear is not distributed evenly over the cross section. Summarizing, then,

$$1.5\,\tau \leqq .57\ S\ (\text{ALLOWABLE})$$

or

$$\tau \leqq .38\ S\ (\text{ALLOWABLE})$$

As for the bending stress on the locking lugs, it is assumed that the breech thrust force is uniformly distributed, so that the lever arm of the bending moment is taken as half of the height of the locking lugs.

Under these assumptions, the following values were determined:

	M60/M14	CARBINE
Service Pressure	50,000 psi	34,200 psi
Cartridge I. D.	.41 in	.31 in
Force	6,600 lb.	2,600 lb.

Resultant stresses are:

	M60	M14	CARBINE Left Lug	CARBINE Right Lu
Shear stress	12,700	31,000	17,500	17,000
Bearing stress	70,000	64,000	90,000	22,000
Bending stress	7,800	50,000	10,000	40,000
Combined stress	23,400	73,400	21,000	50,000

The safety factor is a ratio of the dynamic yield stress to the above stresses. The dynamic yield stress is assumed to be equal to the static tensile strength of the material.

The static tensile strength is:

'S' TENSILE = 168,000 psi for M60 & M14
'S' TENSILE = 172,00 psi for Carbine

Safety factors resulting:

	M60	M14	CARBINE
Shear	5.3	2.2	3.9/4.0
Bearing	2.4	2.6	1.9/7.8
Bending	21.4	3.4	17.2/4.3
Combined	7.3	2.3	8.2/3.4

Therefore, as a guideline, the preliminary average factors of safety for design is recommended as follows:

Shear	3.0*	Minimum
Bearing	2.5	Minimum
Bending	4.0	Minimum
Combined	3.5	Minimum

*Shear stress safety factor $\leq$ .38 static tensile strength

Stresses in Barrels

Much of the small arms barrel strength is designed to resist external forces, such as bayonet thrust loads, pull from sling, grenade launchers and other muzzle attachments, parachute drop, and other adverse or abusive forces.

An excellent arrangement of standard formulae on this subject is given in Pamphlet AMCP-706-252 "Gun Tubes"

However, additional confirmatory data is obtained experimentally, and strain gages are applied to the outside diameter of the barrel in order to measure actual loads (strains) imposed during firing. The internal pressure in terms of the strains measured at the outside diameter is given by the following formula:

$$Pi = 1/2\ E\ \varepsilon_c\ (w^2-1)$$

E = Young's Modulus (30 x 10^6 for steel)

ε_c = Strain in/in measured at O.D.

w = ratio of O. D. to I.D.

For a simple comparative value, the following formula for the maximum tangential stress, which occurs at the inner wall, is frequently useful:

$$\sigma_{tp} = p\ \frac{w^2+1}{w^2-1}$$

w = the wall ratio, Do/Di

As a general rule of thumb, for rifles, the barrel thickness over the chamber should be at least 2/3 of the dia., Di, measured at the midpoint of the cartridge body.

For 7.62mm NATO: dia $=$ .465,
then th. $=$ .310
O. D. $=$ 1.085
$w = 1.085/.465 = 2.34$
$w^2 = 5.45$

Using a proof pressure of 65,000 psi, the resulting maximum tangential stress at that point in the barrel is approximately:

$\sigma_{tp} = 65{,}000 \times 6.45/4.45 = \underline{\underline{94{,}000 \text{ psi}}}$

For machine guns, where temperature stresses are a factor, the wall thickness should be increased so that the maximum tangential stress is in the order of 80,000 psi. Another useful formula is that :

for thin walled cylinders: ($t < 1/10\ D$)

hoop stress $=$ pr/t

axial stress $=$ pr/2t

Stress on Bolt Locking Lugs

In a rifle design, the bolt lugs are usually highly stressed, as noted in the previous chapter on "Factors of Safety".

The Cal. .30 M1 rifle is used as an example, since, as a rotary bolt with front locking lugs, it typifies most of the modern military rifles.

Several methods may be used to determine the load developed aginst the bolt face during firing.

The first shown is in knowing the shape of the pressure-time and pressure-travel interior ballistic curves and equating impulse to momentum, so that

$Ft = mV$

In firing a 150 grain bullet with 50 grains of propellant, and a muzzle velocity of 2700 ft./second.

The peak pressure is 50,000 psi, and acts for .2 msec. before a projectile travel of 1.5 inches with a velocity of 1200 fps causes a decrease of pressure in typical hyperbolic fashion. For this period of projectile acceleration, $F = m\,a = \frac{m\,v}{t}$

$$F = \frac{150 \times 1200}{7000 \times 32.2 \times .0002} = \underline{4000 \text{ lb.}}$$

This is the thrust load acting on the projectile at the time of peak acceleration.

Compare this with the projected peak pressure, acting on the bore diameter and the cartridge case.

$F = SA = 50,000 \times \pi/4 \ .308^2 = 3750$ lb., which compares with the force of 4000 lb.

Inside the cartridge case, the inner dia is approx. .410", therefore

$F = 50,000 \times \pi/4 \ .41^2 = 6,600$ lb.

This may be taken as the peak thrust against the bolt face. The bolt thrust forces are developed by the peak pressure acting across the inner case dia, not the chamber dia. In the event of a case rupture, then the pressure does act across the chamber dia, increasing the bolt thrust considerably. The difference between projectile and bolt thrust is the force acting against the annular case neck, obturating the gases. In this case it is approximately 2850 lb. and acts to stretch the cartridge case in the chamber, requiring headspace control. This will be discussed in detail in the chapter on Headspace. It should be noted that any looseness in headspace will increase the impact effect of the case and bolt against the locking lugs.

In calculating the stress on locking lugs, four stresses are considered:

(1) Bearing

(2) Shear

(3) 45° shear

(4) Bending

(5) Combination of above 4 stresses

Using the .30 cal. M1 example, the bearing surfaces are approximately .081 in.^{2} which results in a direct bearing stress of 5,500 lb./.081, of 68,000 psi. Apply a factor of 1.35 as an impact load factor, due to the nature of dynamic loading, resulting in a stress of 92,000 psi.

Now the direct shear area should be approximately at least twice as long as the direct bearing height. This will provide, in this instance, a shear stress of 46,000 psi. and provide a lug that will

be essentially free of high bending stresses in the axial direction. In the actual M1, the lug is longer than necessary; therefore the shear stress is lower. (31,000 psi) Bending stresses in the radial direction are more complex, and formulae may be found in Roark's "Formulae for Stress and Strain".

The 45° shear stress is known as Lueder's slip lines, and is a plane 45° to the bearing surface. This is the plane of maximum shear stress. In this example, the shear stress from this source is $92000/\sqrt{2}$ or 65,000 psi.

The radial (inward) bending stress depends upon the construction of the bolt body. Hollow shell type bolts, may suffer high stresses in this respect, but a fairly solid bolt head, as in the M1, is rugged in this respect.

It cannot be overemphasized that sharp inside corners, particularly on the stressed corner where the locking lug is developed, should be avoided since a sharp inside corner is the beginning of a crack. On some bolts considerable inletting of grooves are specificially machined in order to avoid the presence of a sharp corner. This is done on the M60 bolt, for example.

Note that in the M1 bolt structure the locking lugs are positioned high with respect to the lateral centerline. This results in a definite longitudinal torque to the entire bolt body, thus positively defining the stress path for each shot. That is, on each round, the upper surface of the bolt body is in tension, the lower surface in compression. This geometry is necessary, so that the feed lips may be raised to a favorable position for feeding.

One common formula for combined stress is:

$$\text{Max. } Sp = \sqrt{(1/2\ S)^2 + Ss^2} = 79{,}000 \text{ psi}$$

Sp = Max shear stress

S = Bearing Stress = 68,000 X 1.35 = 92,000 psi

Ss = 65,000 psi = Shear Stress at 45°

The principal stress, then is

$$S^1 = 1/2\ S \pm \sqrt{(1/2\ S)^2 + Ss^2} = 125{,}000 \text{ psi}$$

This is why points of stress concentration should be avoided, as a plague. Note also that the peak loading time of .2 milliseconds develops a safety factor, since the bolt lugs undoubtedly react to a lower average load.

However, the 125,000 psi combined stress value is a good reference point since the yield strength of this material is in the order of 160,000 psi.

There are a number of methods by which a bolt lock may function. This includes rotary bolts, tilting bolts or locks, propped locks, wedge locks, ad infinitum. However, each good locking system should have a number of distinct characteristics that enable it to function efficiently with the high pressure characteristics of the cartridge case. The most prominent mechanical design feature is that commonly referred to as "primary extraction" or "initial extraction".

It is that characteristic in design that enables the cartridge to slip rearward only several thousandths of an inch in the chamber prior to unlocking and full extraction. In this way, the cartridge case is pried away from its sealed-in position in the chamber, and some semblance of an acceleration to the cartridge is applied. At this time of initial extraction the chamber pressure is still in the order of 700 - 1000 psi, therefore, the cartridge case is still in a condition of stress, particularly at a point just to the rear of the shoulder, where the wall thickness is near a minimum. This is where case stretching occurs, and will be discussed in further detail in the chapter on "headspace".

However, the bolt locking lugs, the subject of this topic, should provide initial extraction because, for rotary bolts, the lug bearing surface is machined at a helix angle in the order of 4°. Angles larger than 4° are not recommended, as the bolt lugs will not be self-locking, but will tend to transmit a rotary pulse upon firing, therefore battering the locking cam and/or stud. This helix angle not only cams the cartridge into the chamber but also helps back out the cartridge case, unseating it from the chamber walls preparatory to the formal extraction stroke and this distance is equal to the arc length rotated by the base of the locking lug times the tangent of 4°. Therefore, rotary bolts with large numbers of small lugs are limited in this extraction distance because of the small angle or rotation.

As a simple experiment in observing the self-locking stability of a bolt, merely assemble the bolt into the locked position without the operating rod or bolt carrier, and with the gun held vertically, muzzle upward. Carefully drop a long, brass rod down the bore, so that it will impact sharply on the bolt face. While the bolt will prance up and down, if the locking surfaces are self locking, the bolt will not rotate and fall out. Repeat this several times, particularly if the bolt creeps in rotation. Bolts that tilt, as the cal. .50 M8 Spotting Rifle and the 7.62mm FN rifle, or have tilting locks, as the B A R, also exhibit primary extraction. As the bolt locking surface rides up the locking abutment during unlocking, it will be observed that the bolt face moves rearwardly a proportionately small distance. The same is true of the Browning Machine Guns, in which the locking face is not vertical, but inclined rearwardly approximately 3° - 4°.

The M73 machine gun has a cross-sliding bolt that moves, not perpendicular to the bore, but at a receding angle of 4°. Again this cams the cartridge into the chamber, during feeding, and pries the spent case out during initial extraction.

The cam driven, externally powered weapons, such as the M61, M75, M129, and XM140 have this primary extraction feature built in to a marked degree because of the low slope of the initial opening phase of the cam.

In addition, it may be noted that most of the modern military and commercial weapons employ front locking lugs on the bolt. This shortens the amount of breech deflection at the instant of peak pressure, thereby improving headspace control.

A practical note may be added here in that the bolt should have an adequate impact surface against the barrel or breech face when the operating rod is released without ammunition being fed, that is, an empty weapon. This impacting will occur frequently, therefore adequate bearing surface should be provided in order to prevent peening or jamming.

Firing Mechanism Design

The starting point for designing the elements of a firing mechanism is a consideration of the primer sensitivity, or the energy required to fire 100% of the primers 100% of the time. A large number of misfires will result if the striker blow is less than the maximum of the drop test specification for that primer. For example, in testing primers, a sampling of .05% is taken (one in 2000) so that, statistically, if one primer of 1000 samples misfired at the high drop level, the chances are about one in five of obtaining misfires at this energy level of a group of 200 primers taken from this lot.

Inspection methods are usually required to insure that a given small arms weapon has adequate striker energy delivered to the firing pin. This may be done by the firing pin indenting a standard copper crusher gage when struck. A test fixture is designed in which the primer is replaced by a copper cylinder.

Coordination is required between ammunition and weapon designers as to the accepted level of primer indent energy required for each system. Then, the striker, or firing mechanism energy is designed for twice (or 1.5 as a minimum) the specified "all must fire" primer strike energy.

The following standard primer drop test specifications are summarized:

CALIBER	WGT. OF BALL	HEIGHT AT WHICH ALL MUST FIRE	ENERGY
.30 Rifle	4 oz.	15. in.	3.75 in. lb.
.30 Carbine	2 oz.	18. in.	2.25 in. lb.
.45 Cal.	2 oz.	16. in.	2.0 in. lb.
.50 Cal.	8 oz.	17. in.	8.5 in. lb.

An indent test of various weapons (using copper pressure cylinders) was performed, and could be made part of an automatic burst, since the instrumented cartridge case could be loaded in a clip or belt.

A. Correlation with drop test (4 oz.)

DROP HEIGHT	INDENT
9"	.013 in.
12"	.015 in.
15" (All fire spec.)	.017 in.

B. Rifles

M1903	.020 in.
M1917	.017 in.
M 1	.016 in.
BAR - first shot	.014 in.
slow automatic	.016 in.
full automatic	.013 in.

C. Machine Guns

M1917 - first shot	.015 in.
full auto	.023 in.
M2 AC - first shot	.017
full auto	.020 in.

M1918 A4 first shot	.018 in.
full auto	.021

Therefore, a number of weapons appear to be marginally close to the 15" drop specification.

To better understand primer test procedure the following is repeated from a specification for the M42 (Fuze) primer, in which the drop-weight is a steel ball weighing 1.94 ± .02 ounces.

Procedure of Test

Drop test 50 primers at 8 inches. Then drop test 50 more primers at 9 inches, at 10 inches, etc., until a height is reached at which all the sample of 50 fire. If the test is run to 13 inches and misfires still occur, reject the lot. Then for all other lots, test 50 primers at 7 inches, 6 inches, etc. until a height has been reached at which none of the samples fire. If the test is run to 2", and fires still occur, reject the lot. The test procedure is terminated as soon as a height at which none fire has been reached"

When the striker, firing pin, or hammer hits the primer, the primer pellet is compressed between the anvil and the firing pin tip. This blow must be sharp, so that the firing pin energy will not be dissipated, causing misfires. The firing pin tip should not be too large in diameter or the resulting pressure force will shear the primer cup into the firing pin recess, either plugging the hole or severely eroding the breech face by high pressure gas leakage.

The primer strike is one end of the firing mechanism, while the other end, which starts the train of events is the trigger pull. Here is where a firing mechanism is so unique with respect to all other mechanisms and devices. The trigger pull must be a smooth, carefully gauged motion, or "feel". Trigger pull is in the order of 8 lb. ± 2 lb. and the trigger pull is usually taken in two phases, a slack, and a squeeze. The mechanical motion must be smooth, and not gritty, so that the marksman does not know when the hammer is released. If he did know, he would flinch, quite involuntary, and thus miss his shot. If, at any time during the trigger pull, the shooter changes his mind and decides not to shoot, the linkages should return to their starting position, and not be "hung up" partway along the sear surfaces.

For purposes of demonstration, the M14 firing mechanism, derived from the M1 Rifle, shall be discussed in detail. Mechanically, if contains many ideal features, as well as all of the required elements and characteristics of any typical firing mechanism.

Outwardly, it is compact, modular, rugged, foolproof in assembly, contains few components, and is accessible for cleaning and inspection. It contians one coil spring and one formed wire spring; and two cross pins, both of which are arrested from cross motion during firing, by the receiver.

The safety lever is ideal, in that its position is instantly recognized by sight or feel. Further, once the safety is applied, it is not easily unsafed by accident. The trigger finger "safes" or "unsafes" the weapon, minimizing motion of the shooter's hand or arm.

The components, or elements, required in a typical firing mechanism, and as contained in the M14 firing mechanism, are as follows:

1. Housing - This is the form of a three-walled box-type structure, and is highly rigid, supporting the lower receiver bridge and magazine housing. In conjunction with the trigger guard, it clamps the stock tightly to the receiver. This is important, in that it eliminates looseness in the housing, receiver, and stock groups.

2. Trigger Guard - Made of sheet metal, it cam-locks the housing to the receiver, clamping the stock up tightly. It is formed so as to be spring-tight in the latched position, with a bullet tip hole provided for use with tight samples. As the trigger guard is swung open, the hammer is automatically cocked.

3. Trigger - Simple, rugged, and comfortable to the finger. The spring moment-arm is extremely short, in order to reduce loads to an acceptable level. After hammer release, finger follow-through motion is short and comfortable.

4. Primary Sear - In the M14 mechanism, this is integral with the trigger. In many other mechanisms, it is the component that the trigger moves to release the hammer. The hammer "claw", "hook" or "notch" must work at a slight pulling angle when the primary sear is in motion. In this way, the hammer/sear will not slip or creep due to vibration, but kinematically is self-holding.

5. Secondary Sear - This is the component that engages the hammer during the cycle of operations after firing while the trigger is still pulled. The secondary sear should hold the hammer in a lower position than the primary sear. When pressure on the trigger is relaxed, control of the hammer then passes from the secondary sear to the primary sear. During trigger pull, the primary sear moves partway from the hammer claw in the "slack" phase. Then the back of the hammer claw bears against the secondary sear for increased pressure in the "squeeze" phase. An extension on the side of the secondary sear provides a bearing surface for the sear release, in full automatic fire.

6. Sear Release - Mounted on receiver, in the M14 rifle. In automatic fire, this component functions to release the hammer upon the timely traverse of the operating rod. That is, the hammer is held until the bolt has been closed and positively locked by the operating rod.

7. Connector - Mounted on receiver. In automatic fire, this component is actuated by the operating rod to rotate the sear release, firing the weapon. The timing of release is a function of operating rod position. (Bolt is locked, bounce under control)

8. Safety - The safety locks the hammer and blocks the trigger. It cannot be engaged unless the hammer is cocked. This is important, as a firing mechanism can be broken otherwise. That is, if a safety mechanism is first engaged, then the hammer cocked, then the cocking motion would cause the hammer to interfere with the safety. As the safety engages the hammer, it cams the hammer lower than the primary or secondary sear. This is necessary for tolerancing purposes so that, when the safety is released, the primary sear will always be in position to re-engage the hammer.

9. Hammer - In this mechanism, the hammer can be repeatedly released in "dry firing" without breakage. The striking face is accompanied by an extension that engages the base of the bolt, camming it closed so that the weapon cannot fire in an unlocked position. This is important. The left side of the hammer also contains an integral stop lug, so that the hammer is positively stopped during inertia travel, preventing breakage of parts. (cocking) Note that the position of the hammer claws is far removed from the hammer pivot point, thereby minimizing loads on the hardened gripping surfaces, thus reducing chipping, galling, and gritty trigger pull.

Note that the point of application of the hammer spring is such that the maximum spring leverage is applied when the hammer is striking the firing pin. That is, as the hammer rotates to a cocked position, the lever arm rotates to a lesser value, thus reducing cocking forces.

The hammer energy may be taken directly as the spring energy, or FXS; that is, mean spring force times spring stroke.

Time for hammer fall or firing pin strike is given by the formula:

$$t = \sqrt{\frac{2WS}{gf}}$$

s = travel (c.g.)
g = 32.16
F = avg. spring force
W = Wgt. striker + 1/2 spring

If at all possible, a rotating hammer should strike the firing pin at the center of percussion of the hammer. This is the theoretical point through which the line of action passes of the resultant of all forces acting on the body. It is given by the formula:

$$\ell = Jm/xoM$$

where

ℓ = distance (ft) from axis of rotation to center of percussion

Jm = mass moment of inertia (lb./g - ft^2)

m = mass (lb./g)

Xo = distance (ft) between axis of rotation to center of gravity

The center of percussion is correctly developed by re-distributing hammer mass.

10. Hammer Plunger - a headed pin that bears against the hammer notch.

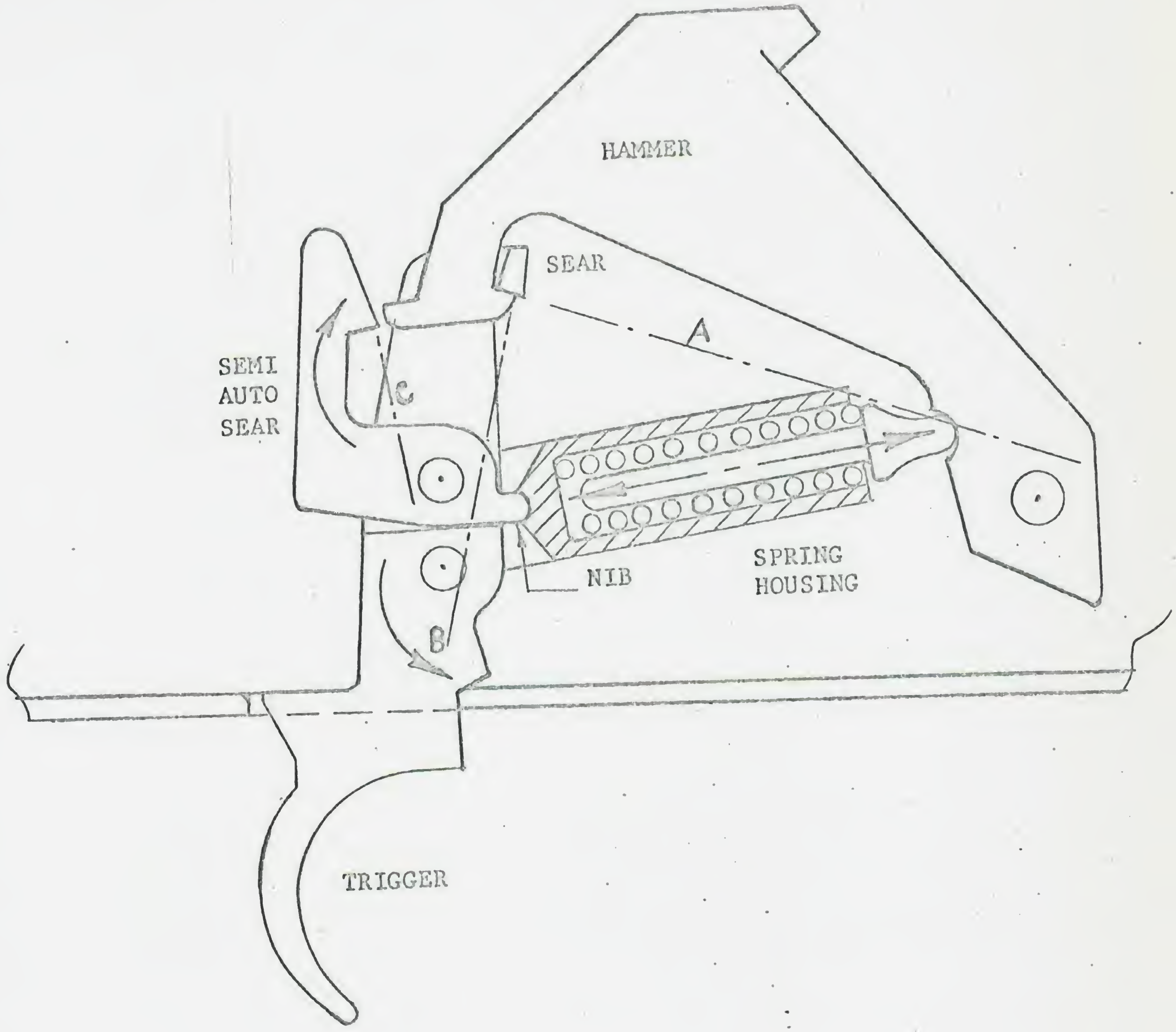
11. Hammer Spring - a plain coil spring. Spring design will be discussed in detail in the chapter on "Spring Design".

12. Safety Spring - a formed wire spring that firmly latches the safety in either the safe or fire position. Note that the bottom leg of this wire form, in the M1 Rifle, also functioned as the clip ejector spring.

13. Housing, Spring - This component is simple in appearance but performs a delicate function, that of directing the spring force on the tips of the secondary sear bits. Looking back at the trigger assembly, a force on these bits produces a short counter-clockwise moment on the trigger, and a clockwise moment on the secondary sear.

To ensure that the bits engage properly for each assembly, the housing is slotted on one side, so as to fit around the safety, thus preventing reversed assembly. Note the two small holes that provide seats for the sear bits.

In summarizing the M14 firing mechanism, it is dis-assembled only with the bullet tip, and re-assembled without any tools. It permits dirt, sand, etc., to work through the mechanism without any inside pockets for same to collect in it.



M 14/M1 TRIGGER GEOMETRY

Note that the position and engagement angles between the hammer claw and sear surfaces is a delicate balance between a self locking grip and trigger pull. The alignment plane of the sear and hammer hook, A extended longitudinally is just above the hammer pivot point. If it were below, there would be a tendency of the hammer to slip off.

A perpendicular plane to this surface, B , comes just forward of the trigger pivot point, giving the trigger a counter-clockwise moment, as required. When the secondary sear is engaged, a perpendicular to this surface, C , is just to the rear of the sear's pivot point, giving it a clockwise moment, as required for a self-holding grip.

Self-styled marksmen at times alter their sear surfaces in order to lighten trigger pull. T H I S I S A D A N G E R O U S P R A C T I C E, as the slope could be transformed to a creeping self-release angle, which would be extremely dangerous.

The M16 rifle firing mechanism also has all of the required functions. If the bolt is held unlocked, the bolt carrier would hold the hammer head back from falling prematurely. The hammer has three distinct notches, in different locations, for the trigger sear, the secondary sear, and the full automatic sear. The secondary sear, here called the disconnect, is unlatched by the trigger being returned to rest, while the trigger sear enters its own notch. The automatic sear is released by the bolt carrier.

At times a firing mechanism is required to perform additional functions, such as in the SPIW program, where an automatic burst counter was specified. In this, the weapon would fire three rounds and automatically stop. A counter mechanism was incorporated which would automatically trip off the sear when the third segment of its travel was executed.

Then it would re-cycle itself for the next three-round burst. If the selector was then set for semi-automatic fire, and only one or two rounds fired, then the mechanism re-selected to controlled burst fire, and the counter wheel has to discriminate this fact and re-set itself.

Machine gun firing mechanisms are quite simple, usually requiring only a primary sear and a safety. Trigger squeeze is not a significant factor, and the minimum of parts insures ease of maintenance. The most critical problem may be in sear bounce, causing run-away firing. This may be resolved by shaping the notch to a self-locking angle of approximately 5^{o}, and permitting adequate dwell travel in the notch length so that the sear has sufficient time to assert itself.

TABLE OF PRIMER ENERGY

TYPE	FT - LBS. *
Small Pistol 500	5.5
Small Piston Magnum 550	8.8
Small Rifle 400	6.0
Small Rifle Magnum 450	7.2
Large Pistol 300	7.1
Large Pistol Magnum 350	9.15
Large Rifle 200	9.2
Large Rifle Magnum 250	9.45

* Energy imparted to a piston in a test cylinder.

A uniform striker energy of .45 ft. - lb. was used for all primers.

The pressure in the cartridge case is raised several thousand psi by this primer energy.

Feeding

Feeding is the action of placing each cartridge, in turn, into the receiver at a position in back of the chamber. The forward, or counter--recoil motion of the bolt then pushes the cartridge into the barrel for firing. This very simple process is the most critical in the design and development of a weapon, as "ff" (Failure to Feed) is the most frequent type of stoppage encountered in the development testing of an automatic weapon. This is because there usually is a portion of the feed process in which the cartridge is not controlled. This may occur at the following points of the feed cycle:

A. For a magazine feed automatic rifle:

(1) Bolt over-ride may occur because the spring-biased stack of rounds was surging and the base of the top round bounced below its pick-up position.

(2) In pushing the top cartridge forward, the cartridge rim rides along under the magazine lip. At the end of this lip, the cartridge base is suddenly released upward. If the cartridge is not controlled in the barrel mouth, receiver, and/or bolt face, a jam may occur.

B. for a belt-fed machine gun:

(1) The leading linked round can bounce about in the feed tray, unless it is controlled, usually by depressor pawls and a holding pawl.

(2) At the point where the round is freed of the (push-through) type of link, the round base is subjected to an uncontrolled transverse force.

(3) When the cartridge is being pushed between the feed tray and the chamber, it may be free to bounce laterally, so that the bullet tip can jam against an inside contour or corner, jamming the action in a half-open position.

C. Sprocket-fed machine guns:

(1) The linked round entering the feeder should be controlled so that it does not bind upon entering the sprockets. This is because of some tendency of the belt to "fishtail". Control is facilitated by pulling on the center-of-gravity of the linked round.

(2) The cartridge, in being "passed off" from sprocket to bolt face is usually held in a momentary "finger-tip" position, and may be subject to "bounce" caused by vibration.

These "danger zones" are aggravated by external vibrations of the vehicle, such as truck, tank, helicopter, fixed wing aircraft, tripod mount, or human shoulder. The importance of maintaining mechanical control over every phase of the cartridge traverse cannot be over-emphasized, if a low stoppage rate is desired.

Usually, the cartridge feed stroke into the chamber is complicated by the fact that the mouth of the magazine or feedway is not in line with the bore, so that the cartridge must be fed in compound directions. In this case a ramp is constructed between the feed tray or magazine and the chamber.

For best results, the cartridge should be fairly close to the chamber, and the ramp angle, particularly for magazines, should not exceed 40°. Machine guns are usually found to have longer

ramp distances, so the ramp slope is shallower. A series of layouts should depict, in small increments, the cartridge motion. In doing this, avoid any inside corners in the receiver or barrel extension in which the bullet tip may stub if the bullet deviates from the predicted path.

In feeding a machine gun, the cartridge is usually placed in the feed tray in the vertical axis of the bore, but in a rifle, because of the double stack magazine, the cartridges are alternatively to the left and right center. Layouts of bullet ramps from magazine to chamber should reflect this compound angle.

The starting point for a weapon design is in placing the cartridge in a favorable position for feed. This affects the bolt design, bolt lugs, barrel extension, and receiver. Too often the locking system is designed first, then the feed system designed afterward. They should be integrated. The radial distance from bore to cartridge should be minimized. This is why multiple chamber revolver guns are more reliable: because the cartridge is already in line with the bore for the critical ramming step.

The Browning and Maxim families of machine guns draw the cartridge rearwardly out of the belt in recoil and transfer the cartridge to the bore axis in c'recoil. This places a greater demand upon the feed system, as it requires that the round is already in belt lead position during the recoil stroke. As a result, the belt feed stroke must occur during the counter-recoil motion. Belt feed is the advancing of the entire belt (or portion thereof) one pitch distance. The feeding of the belt is the portion of the weapon cycle that requires the greatest amount of energy to perform efficiently. Paradoxically, the weapon has more energy in its recoil motion than in counter-recoil, therefore, available energy for the Browning or Maxim system is limited. In the M60 machine gun, the belt feed occurs during recoil motion of the bolt.

Revolver-type machine guns such as the M61 and XM134 have continuous motion feeders, as does the XM140, which is ideal in this respect: <u>The Feed System Should Be Designed To Feed During A Maximum Portion Of The Weapon Cycle</u>.

Since the belt feed requires the greatest amount of work in the weapon cycle, this work should be distributed over the widest time frame, in order to reduce peak loads.

In Chinn's "Machine Gun" Vol. IV, a wide variety of feed mechanisms is shown in an excellent array of sketches. However, study only those mechanisms that were in an actual weapon. (See Appendix A.) Those designs designated as someone's "Patent"

should be disregarded, since feasibility is not proven. Before designing a feed system, it is well to review these mechanisms, conveniently displayed as to mechanism group, such as lever type, cam operated, sprocket type, etc. Note that, particularly with the lever type, the belt feeding is accomplished, in many designs during both recoil and counter-recoil motions. A common form is a dual lever (scissors-type) that provides 35% to 50% of the feed during recoil and 50% to 65% during counter-recoil, depending upon design philosophy.

The lever form or cam shape that controls feed velocity should be designed to produce a minimum belt velocity at all times. The ideal form of constant minimum velocity feeder is incorporated in the XM140, XM134 and M61; the feeder operating during 100% of the weapon cycle time. The resultant velocity diagram then is constant velocity. This philosophy is unique to the design of feed cams, and will result in minimizing belt surge, belt loads, and belt separation or breakage. Minimum constant velocity produces a more "even" pull on all of the rounds in the belt. Other cam forms would produce higher loads on subsequent rounds in a belt. That is, during a feed stroke, not all rounds are pulled at once. This varies from shot to shot and depends upon the condition of the belt in, and adjacent to, the feed tray.

Initial acceleration is not as high as supposed, because of the flexibility of the belt, feed levers, cam path backlash, etc.

The link should not be too stiff, then, in order for the linked belt assembly to function somewhat as a shock absorbing spring during any peak loading. The formula for belt load (max effect of acceleration) is given as:

(1)

$$P = W + \left[\left(W/2\right)^2 + \frac{W_T K}{g} V^2 \right]^{1/2}$$

W = Total weight of belt

W_T = Weight of linked rounds being accelerated

K = Spring rate of linked belt (lb./in.)

V = Maximum velocity of belt

g = 32.16

Note how the force, which is the feed mechanism loading factor, increases with Velocity squared, and with K, belt stiffness.

The formula is developed as follows:

Consider the energy imparted to the ammunition belt:

(2) $E = \frac{W_T V^2}{2g} + W(X_d - X_s)$

X_d = dynamic deflection of belt

X_s = static deflection of belt

Also:

(3) $E = K X_d{}^2/2 - K X_s{}^2/2$

therefore:

(4) $\frac{W_T V^2}{2g} + W X_d - W X_s = \frac{K}{2} (X_d{}^2 - X_s{}^2)$

so:

(5) $X_d{}^2 - X_s{}^2 = \frac{W_T V^2}{gK} + \frac{2WX_d}{K} - \frac{2WX_s}{K}$

transposing;

(6) $X_d{}^2 - \frac{2W X_d}{K} = \frac{W_T V^2}{gK} + X_s{}^2 - \frac{2WX_s}{K}$

completing the square:

(7) $X_d{}^2 - \frac{2WX_d}{K} + \left(\frac{W}{K}\right)^2 = \frac{W_T V^2}{gK} + X_s{}^2 - \frac{2WX_s}{K} + \left(\frac{W}{K}\right)^2$

factoring:

(8) $\left(X_d - \frac{W}{K}\right)^2 = \frac{W_T V^2}{gK} + \left(X_s - \frac{W}{K}\right)^2$

Taking the square root:

(9) $X_d - \frac{W}{K} = \sqrt{\frac{W_T V^2}{gK} + \left(X_s - \frac{W}{K}\right)^2}$

Now, by definition, static deflection equals W/2K.

therefore; $X_s = W/2K$

(10) $$X_d = \frac{W}{K} + \sqrt{\frac{W_T V^2}{gK} + \left(-\frac{W}{2K}\right)^2}$$

The load developed is simply the dynamic deflection times the spring rate of the linked round.

(11) $$P = X_d K$$

Therefore, combining the above two formulas:

(12) $$P = W + K\sqrt{\frac{W_T V^2}{gK} + \left(\frac{(-)W}{2K}\right)^2}$$

(13)→(1)

$$P = W + \sqrt{\frac{W_T K V^2}{g} + \left(\frac{W}{2}\right)^2}$$ {WORKING FORMULA

This load should be considered in calculating the strength of every component (the feed mechanism) such as shear strength of feed pawl pivot pin, compressive stress between roller and cam, etc.

The pressure required to dent the cartridge case should also be considered, and may determine the required bearing area between feed pawl and case. (esp. caseless ammo.)

$$P_{cr.} = \frac{E h^3}{4 (1-\mu^2) R^3}$$

E = modulus of elasticity of cartridge brass

h = wall thickness of case

u = Poisson's ratio, = .28

R = outside case diameter at contact

Link Design

The link design is integrated with the feed system, and links are generally classed as side-stripping, push-through, or pull-out, depending upon the type of motion used to separate the round from the link. There is an excellent display of link designs shown in Vol. IV of "Machine Gun" by G. Chinn, page 285.

Links should have the following characteristics:

(1) Pull the cartridge at the linked round center of gravity

(2) Lock onto the cartridge, so as to prevent "walk-off" or a progressive vibrating off its position on the cartridge (axially)

(3) Be tight enough to grip the cartridge so that during maximum belt pull forces, the round does not loosen (radially)

(4) Not be tight enough to prevent easy round stripping in feeding

(5) Belt be flexible in a butt fan radius

(6) Be flexible in a nose fan radius

(7) Be flexible in twist

(8) Be able to fold in parallel layers in an ammo can. (front & back)

(9) Must permit the layers to stack evenly.

(10) Linked belt must not disintegrate when armorer handles belt outside of weapon and chuting (Infantry load)

(11) Links must disintegrate upon ejection from weapon

(12) Linked round must not have protrusions that cause belt to snag in ammo can, chuting, feeder, weapon, or ejection chute

(13) Links must be re-usable (for development test purposes) without significant change in characteristics upon re-use

(a) stripping force

(b) gripping force

(14) Capable of long term storage without change in gripping force and without corroding.

(15) Cost should be low

(16) Adaptable for either left or right hand feed

The link, in pulling on the center of gravity, should have cartridge gripping sections that straddle the grip area, without being loosened as belt pull increases. The gripping force between link and round is calculated by treating the gripper as a cantilever beam extending from a fixed end. Apply the formulae for stress and deflection,

the deflection being the interference between cartridge and free link.

$$S = \frac{Mc}{I} \quad \text{and} \quad \delta = \frac{W\ell^3}{3EI}$$

I = moment of inertia of tab section

This is a good approximation, with more detailed analysis given in Roark's "Stress and Strain" on curved beams.

Some links have been designed with separate "Cap" sections so that the belt pull forces do not tend to spread open the link. The cap is stripped off the link during entry into the feeder.

The M75 weapon utilizes an extremely lightweight link in relation to the cartridge size because the link is a closed loop encircling the cartridge. In this way, belt loads tighten the grip between link and round, rather than loosen it.

The linked belt pitch distance should be as small as possible for maximum efficiency in feeding, yet be long enough to permit parallel stacking of rows. (particularly at the row ends) A short pitch distance means a short stroke of the feed system (levers, cams, etc.) and reduces the weapon profile. For links that pivot on a succeeding cartridge, the pitch distance is approximately 1.3 times the cartridge body dia., while for links that pivot in the hooks between rounds, the pitch distance is approximately 1.4 to 1.45 times the cartridge body diameter.

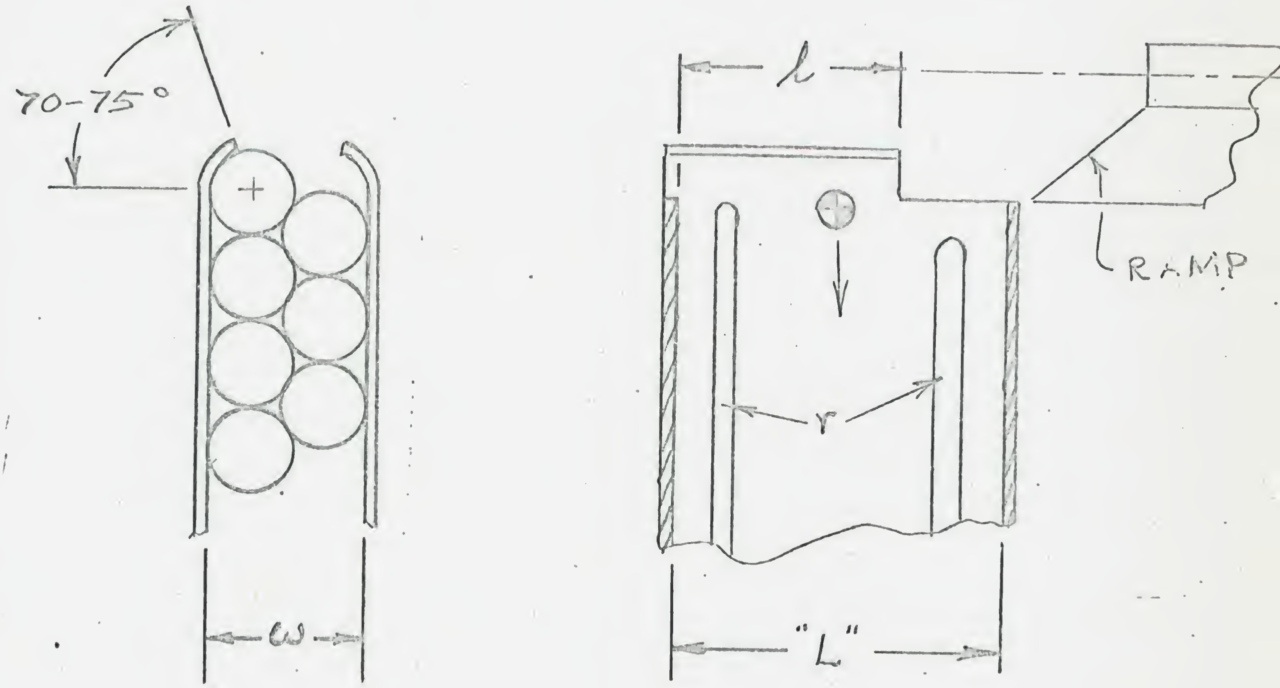
Linkless helical drum type feeders are also employed in some weapons such as the Lewis machine gun (radially stacked drum) and Thompson sub-machine gun (axially stacked drum). Large capacity linkless feed drums are also used (M61 and XM134) which control several thousands of rounds with a conveyor system. This item is a specialized subject of its own.

A recent problem has evolved in which electric primed ammunition becomes over-sensitive in the presence of electronic equipment over a period of time. To prevent the primer from forming an induced electro-magnetic field, a tab of the link is incorporated as an integral shield.

Magazine Design

The magazine unfortunately was conceived as a cheap, throw-away, one-time-use item, and, accordingly is constructed of rather light gauge sheet metal, which is subject to damage in rough handling. Unfortunate, because the magazine lip is one of the most critical surfaces in the entire rifle. Attempts have been made to incorporate a fixed machined lip as an integral part of the receiver in order to eliminate this weak point in the system.

The magazine should be as high as possible into the receiver, commensurate with bolt configuration, in order to optimize cartridge ramping. The magazine feed lips should wrap around the top cartridge so as to prevent cartridge pop-out in handling. This is usually an angle of 70° to 75° as depicted in the following sketch.



The magazine lip length "ℓ" should be long enough to contain the cartridge center of gravity, yet short enough so that the following cartridge, or follower, can give the base an upward moment upon cartridge release, preventing jamming between bolt face and ramp area.

The inside magazine width,"W"for a double stack, should be approximately equal to the cartridge diameter plus cosine 30° times cartridge diameter. If the width is greater than this circle contact, then greater forces are transmitted to the magazine side wall, causing hard feeding and subsequent binding. If the width is shorter, the magazine depth becomes unnecessarily longer.

The magazine side walls should have longitudinal ribs "r" as a cartridge bearing surface to permit easier feeding. Sand, grit, and other matter will then collect harmlessly in the void between the ribs. Secondly, the ribs reinforce the sheet metal wall, greatly increasing its section modulus against deflection.

Inside length "L" should be minimized in order to reduce impact of the bullet tip during automatic fire.

In summary, when designing a magazine, first study other successful magazines in detail, in order that all required characteristics are incorporated. Magazine spring design will be discussed in the chapter on "Spring Design".

Magazine depth must be carefully controlled. This should not be so from an engineering viewpoint, but for practical reasons is necessary. For example, in a 20 round magazine, if the user can squeeze in a twenty-first round, he will do so. Then, when this is inserted in the rifle it may cause hard stripping, or, in bearing tightly against the bolt, may cause a short recoil, or may spread the magazine sidewalls, deforming it enough to cause subsequent jams. The accumulated tolerances of max/min cartridges plus magazine tube tolerances are involved.

The magazine follower and spring is usually designed so that the load is concentrated approx. 35% of "L" from the cartridge base. This is to prevent cartridge base drop during surging of the stack. Also, it helps in lifting the full stack properly, since the bottom cartridge will be inclined approx 15° due to body taper accumulation. For a thirty round magazine, the accumulated incline is so high that the magazine body usually is curved to compensate for this inclination.

<u>Ammunition Storage and Boosters</u>

Weapon systems on aircraft usually have such a large ammunition supply, or such a long, torturous path from ammunition can to weapon, that the weapon feed mechanism cannot pull the load without reducing firing rate or causing excessive wear of cam surfaces and components. For this reason, boosters are used; that is, auxiliary feed devices usually in the form of a motor driven sprocket.

The main problem is in correlating booster delivery with weapon rate of fire. This is not simple, since weapon rate of fire is not constant, hence a sensing device must be incorporated that will control the booster speed and delivery.

A. One method is to used a mechanical pitch sensor. When ammunition is bunched up between booster and gun, the belt pitch will shorten and a mechanical device will stop the booster. As ammunition is used, the belt will stretch out and increase pitch, so that the sensor switches the booster on. The response has to be quick, because as the pitch is increasing, the weapon is firing; the slow inertia of the booster motor and drive may not supply the next round that the gun demands. Consequently, belt pull will be high.

B. The same approach as above, but based upon belt catenary. Catenary is the droop of a length of belting causing a switch to close, controlling the booster.

C. A dual speed booster, where the sensor switch does not shut off the booster, but switches in a resistor that slows the booster slightly. Here response time is quick, because the booster does not have to build up speed from a zero start.

D. Time delay may be a critical factor, as in intermittent fire, when firing stops, the booster has sufficient inertia to feed a fraction of a round more. A few spasmodic bursts will soon "overfill" the feed tray causing a gun jam. A short-time delay control is incorporated in the booster circuit on situations of this type in order to clear the pile-up.

E. A round counter mounted on the feeder that pulses a feed station on the booster sprocket.

Spring Design

In designing a spring, usually a helical compression spring, for some part of a weapon mechanism, it usually happens that only a minimum of space is left for the spring; that is, if the spring is not carefully designed, it will be vulnerable being over-stressed in service.

Firstly, the mechanical requirements of each component, then the strength and durability levels must be satisfied. Secondly, the load requirements of the spring are specified, and, finally, the amount of spring stroke is determined. For the reason that the stress levels of weapon mechanism springs are high, the material to be used is music wire, which has elastic limit stress values of 150,000 to 180,000, depending upon the wire diameter. The smaller the wire size, the greater the permissible yield stress. Springs subjected to high temperature levels are usually stainless steel, such as "Elgiloy". Common spring steels are reliable when stressed up to 80,000 psi at temperatures of 350°F - 400°F. or less.

The use of square, rectangular, or other than round, wire shapes should be avoided for several reasons. Firstly, the stock is not readily available to most vendors, therefore the cost will be higher. Secondly, these special shapes are not produced in tonnage, as compared with round wire, hence have not had the refining development which has been given to round wire. The resultant yield strength is not equal to that of good grades of round wire.

In specifying spring wire, a number of standard gauge series are commonly used, therefore, in order to avoid error, the wire size should be specified in decimals.

The starting point for designing a coil spring is to determine the outside wire diameter permissible, commensurate with available space, and the minimum operating height of the coil. Then, calculate the load and stress levels of a sample coil that will have a D/d ratio of 6.5 to one, and a number of coils that permit only 3 to 5 thousandths inch between coils. "D" is the mean coil diameter, while "d" is the wire size. The compression of a coil spring is technically, not "compression", but "torsion". That is, the coil of wire is being subjected to a torsional, or twisting, force. As the ratio of D/d decreases, an additional shear stress acts, and this becomes dis-proportionally higher, as the coil becomes tighter. A D/d ratio of 6.5 is quite satisfactory, but this may be adjusted between 5.5 and 7.5, depending upon load desired and resulting stress. In no case should the D/d ratio be lower than 4. The component of additional shear stress due to curvature is known as the "Wahl" factor, and is well covered in any standard work on spring design.

Typical values of "K", the Wahl factor, are:

D/d	"K"	"R"
9.	1.15	1.07
7.	1.21	1.1
6..	1.25	1.12
5.	1.32	1.16
4.	1.4	1.2
3.5	1.47	1.23

The calculated spring stress is multiplied by "K". However, in gun design, a spring does not cycle for hundreds of thousands of cycles, therefore, the level of fatigue does not approach that of other fields, such as valve, automotive, etc. Therefore the factor "R" is more realistic, being midway between 1.0 and "K".

The basic formula for stress in torsion for round wire is:

$$S = \frac{Mc}{I}$$

in which M = torsional moment $= PR = \frac{PD}{2}$

$C = d/2$

$I =$ polar moment of inertia $= \frac{\pi d^4}{32}$

so $S = \frac{8\,PD}{\pi d^3}$

However, this static stress is not a positive indicator of the dynamic performance of the spring. Impact loading can cause a much higher set in springs than solid height stress. A testing machine with the same deformation velocity is required.

The other principal spring design formula is that for rate of deflection, or spring rate

$$R = \frac{G\ d^4}{8ND^3} \text{ lb./in.}$$

G = torsional modulus of rigidity, of steel wire, or 11.5 X 10^6

Note that the Load/Stress formula does not contain a factor for number of coils. This is because the coils are in series and the load required to deflect one coil to solid height is the same required to deflect the entire spring to solid height.

In designing a spring, the formulae for stress and rate are worked together. For example, if a lower rate spring is desired, either "d" is decreased or "D" is increased. This, in turn, reduces the peak load for a given stress, each to a varying degree. Use or a commercial "slide rule" type of spring calculator permits rapid trials of various combinations.

The spring rate should be as low as reasonably attainable. One reason is that stress and stress range govern the life of springs. The wider the stress range, the quicker the spring will fatigue. Comparatively high stresses can be used where the working range is short, and reducing the spring rate accomplishes this. Spring rate is reduced by maximizing the number of coils. This, in addition, reduces the deflection range of coils during spring surge. That is the free distance that individual coils of wire can deflect. For this reason, the spacing between coils, at minimum operating height, is .004 inch, and may even be reduced to .002 inch.

The factor "N" is the active number of coils. That is, springs with squared, or squared and ground ends have one inactive coil on each end.

Springs are generally wound right hand, but when one spring is inside another, they are wound opposite hand, in order to prevent clashing, or pinching, of coils.

As a spring is compressed, the outside diameter increases, due to the closing of the coils. The formula for calulating diameter increase is:

$$D_1 = 1/2\sqrt{P^2 + 4\ D^2 - d^2}$$

D = mean dia.
D_1 = mean dia., solid height
P = pitch at free length
d = wire dia.

The spring natural frequency should also be checked, and should be at least 11 to 13 times higher than the weapon rate. If not, the

spring will surge and the stresses greatly augmented. Some surging can be absorbed by having a few closed coils built-in.

One general formula for determining spring frequency, n, is: $n = 826{,}500\ d/ND^2$ cycles/min.

The surge wave of a spring, when loaded, requires a specific time to travel from one end to another. Velocity is constant for a particular spring and does not depend on the load or velocity of the load. However the magnitude of the wave is affected by the velocity at which it is struck.

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The equation for the surge-wave velocity "C" in a compression spring is:

$$C = \frac{d}{D} \sqrt{\frac{G\ 2}{2\gamma}} \quad \text{(in./sec.)}$$

d = wire dia.
D = mean Coil dia.
$G = 11.5 \times 10^6$
$\gamma = .283$ lb./in.^{3}

thus $C = 88{,}600\ (d/D)$ in./sec.

the surge time T:

$$T = \frac{3.17\ D\ N}{C} \text{ sec.}$$

or $$T = \frac{N\ D^2}{27{,}900\ d} \text{ sec.}$$

When the ratio of the load vs. the spring weight is reduced to 4/1, then the spring weight begins to affect the calculation for energy stored in the spring. The load is then modified by adding 1/3 the weight of the spring.

The limiting factor in the velocity that a spring can drive a component is not the energy stored, but the rate at which this can propagate down the turns of the spring. This velocity is related to speed of sound in the spring material and is limited by the allowable stress in the material.

This ultimate velocity is:

$$V \max = \frac{S}{131} \text{ in./sec. (steel)}$$

This is about 115 ft/sec.

At times, the behavior of some springs cannot be predicted, but can be observed by using a stroboscope.

Of course, proper heat treatment is necessary, and modern equipment in most facilities assures this, but if springs are over-heated grain structure will be coarse and the fatigue life poor. One simple method of determining proper spring design and heat treatment is to close a spring to solid height three times and measure the free length. Any "set", or permanent deformation, should then be apparent.

Compression springs must be guided, either by means of a rod inside the coil, or the coil inside a hole, if the free spring length is 4 or more times greater than the mean coil diameter. Otherwise buckling would be likely to occur.

When dual (inner and outer) springs are used, one spring is wound right-handed while the other is wound left-handed, as mentioned previously, but the outer spring should be designed to carry approx. 2/3 the load, the inner 1/3.

When designing odd shape compression springs, such as a magazine tube spring, two methods may be used. The first, is to take each segment of a single coil and treat that as a cantilever beam with a flexible support, and summing the load and deflections for the four segments (of a rectangular coil) of the single helix. The load will be the same for the total spring, with the rate varying inversely as the number of coils. This method is tedious, but a simpler method is to add the total circumference of one coil of wire and convert that to a round coil of the same circumference. Use the same wire dia., number of coils, etc, and solve for loads and stresses. This result will be within 5 - 10% of actual loads and stresses.

Extension Springs

In designing extension springs, the factors are the same as in the design of compression springs except that extension springs can be wound tightly with an initial tension between the coils so that a load must be applied to separate them.

The types of ends vary widely, depending upon loop, hook, or end desired.

Extension springs are not usually found on weapon mechanisms, because the stress concentration where the end coil is turned to form the loop, or hook, is vulnerable to breakage. Also, for a very practical reason, when an extension spring is being assembled and/or disassembled, there is no positive stop as there is in compression springs (solid height) and thus an extension spring can be very easily over stressed in handling.

A special form of a compression spring is a so-called "garter spring" in which a close-coiled spring is assembled into the form of a ring. This was done in a rifle grenade launcher where the spring acted to retain the grenade. But after it was set into position, a twist buckled all the spring coils so that they locked the grenade tube to the muzzle device. This caused a serious accident, which pointed out how dangerous that type of spring could be in practice.

Torsion Springs

A torsion spring can also be quite useful in weapon mechanisms, being a coil of wire subjected to torque, that is a wind-up of the coil, as in a common "rat-trap" spring. The ends may be configured in a variety of styles, and this is usually useful in reaching out to remote distances to perform load, or retaining, functions.

It is most useful in rotating components, or to cushion shock on rotating parts.

A torsion spring should always be actuated in a direction that would tend to "wind up" the coil, or reducing the diameter of the coil. Otherwise, the end coils would tend to bend outward, carrying all the load. Also bear in mind that when a torsion spring is loaded, its coil length increases, as one full turn makes the coil one wire diameter longer. Allow clearance for this increase in length or the coil will bind or break. The spring leg will also shorten, and if not contained, will snap free.

There is almost no limit to the possible configurations of a torsion spring, but bear in mind that designing shapes beyond those commonly tooled increases cost astronomically.

The specifications for a torsion spring should include the load as a torque in inch-pounds, as well as the mechanical dimensions, including right or left hand coiling. If possible, the spring should be reversible in assembly.

Two formulae required to develop a torsion spring are as follows:
(1) Equation for deflection (round wire):

$$M = \frac{E\, d^4\, T}{10.8\, D\, N}$$

M = Torque (in.-lb.)
N = Number of coils of wire
T = Number of turns spring is torqued
E = 30×10^6
D = Mean diameter of coil

(2) Equation for stress:

$$S = \frac{32\, M}{\pi d^3}$$

M = Torque (inch - lb.)
d = wire size

As in coil springs the ratio of D/d must also be considered, since the stress for a straight beam must be modified for the curvature of the coil. For torsion springs, the stress correction factor (K) increases sharply as the D/d ratio is reduced below 2.5 to 3.0.

In general, the lower the maximum stress, and the shorter the range of stress between initial and final working positions, the longer the service life.

Flat Springs

At times, limitations in space prohibit the use of coil springs, and a flat "strip" type of spring may be utilized. Flat springs have the added advantage of being able to perform a combination of functions.

Special blanking and forming tools are used to produce this item, and a number of specialty vendors are able to produce these quite economically.

Straight carbon steels are used in these springs, with .70 - .80% carbon lending itself better to sharp bends, while the .90 - 1.05% carbon exhibiting higher elastic limits. In some cases alloy steels are also used. Sharp inside corners should be avoided at all times, with holes punched at the ends of slits.

The controlling factor on the loads of small flat springs is the thickness of the material, as the deflection formula shows the load to be a function of the thickness of the material cubed.

Calculations of flat springs are similar to those of beams, and the comparable form should be selected, such as cantilever with fixed end, free end, etc., or beam with distributed load, or concentrated load, etc.

For example, consider a cantilever beam clamped at one end with a concentrated load at the other.

The formulae for stress and deflection are worked together, such as:

$$S = \frac{Mc}{I} = \frac{6\ PL}{b\ h^2} \qquad h = \text{thickness}, \quad b = \text{width}$$

$$F = \text{deflection} = \frac{PL^3}{3EI} = \frac{4PL^3}{Ebh^3}$$

Of course, stress raisers, such as holes and sharp inside corners, must be minimized, particularly where the base of the spring (max. stress) is bending.

Bends in the spring also affect stress, generally according to the following table:

Inside radius of bend	Stress factors "k"
.5 X metal thichness	2.0
.75 " "	1.66
1.0 " "	1.5
1.25 " "	1.4
1.5 " "	1.32
2.0 " "	1.25
3.0 " "	1.16
4.0 " "	1.12

Theoretically, the shape of a flat spring can be modified to improve its efficiency, such as the uniformly stressed spring and the built up (leaf) spring, but this advantage is offset by production costs.

Belleville Springs

This form of washer-type spring is quite common in gun mechanisms, particularly buffers, and have the advantage of being quite compact, the range of motion being quite short.

A series of empirical curves are used to calculate this type of spring, and may be found in most up-to-date spring design manuals. (Associated Spring Handbook). Otherwise, one would have a formula with 4 variables, resulting in extensive work.

A typical solution of a Belleville Spring configuration is as follows:

$$P = \frac{E f}{(1 - \sigma^2) M a^2} \left[(h - \frac{f}{2}) (h - f) t + t^3 \right]$$

P = load
E = 30×10^6
f = deflection = .004
σ = Poisson's ratio = .3
h = free height minus thickness = .005
t = thickness = .084
O.D. = .730
I.D. = .380
a = 1/2 D.D. = .365
M = Constant taken from chart relating O.D./I.D. ratio to stress constants C_1 and C_2
C_1 = 1.21
C_2 = 1.35
M = .675
$\therefore$ P = 867 lb.

Likewise, the stress equation is:

$$S = \frac{Ef}{(1 - \sigma^2) M a^2} \left[C_1 (h - f/2) + C_2 t \right]$$

$\therefore$ S = 172,000 psi

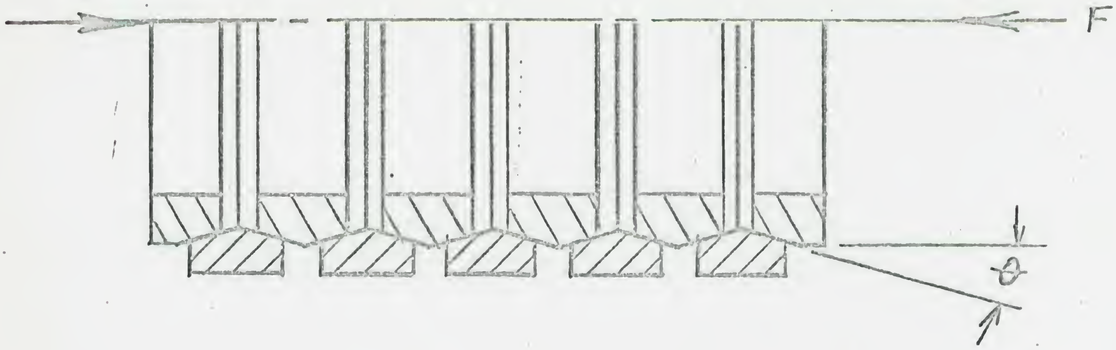
The spring rate of Belleville springs is controlled by specifying the method of stacking and the number of elements.

Belleville spring packages are usually used as buffers, to absorb shock, or to take up slack, somewhat as a spring washer. In a commercial application, Belleville type springs are used as pressure disks for power brakes, so the range of application is wide.

As may be expected the thickness of the plate greatly affects the load characteristics, also the ratio of h/t; that is, the free height vs. the thickness. This ratio affects the load deflection curve, and for certain curves, the springs have a near zero rate for a portion of their travel. Thus, changes in assembled heights (tolerances, wear, etc.,) would not change the load.

Ring Springs

Ring springs have also been used extensively in ordnance, particularly as barrel buffer springs for short-recoil operated weapons. Ring springs consist of a series of inner and outer elements that utilize friction as the means of absorbing load. In compression, the outer elements expand while the inner elements contract, with high frictional loads bearing on the mating surfaces.



Ring springs inherently have a high load capacity for their size and weight and can absorb shock with low recoil.

The spring elements act as inclined planes through the angle θ. The force F is uniformly distributed throughout the circumference of the ring. The resultant force acting on the wedging surface of each sector can be divided into two components, one normal to the surface Fn and the other tangential (friction) equal to Fn times u (coef. of friction)

Charts of ring spring compression constants and recoil constants in terms of taper angles and various coefficients of friction (usually from (.10 to .18) have been prepared and include taper angles of from 10° to 30°.

The axial load is less during unloading than during loading, of course, but the radial loads are the same.

Some recommended proportions for ring spring packages are:

(1) The compressed height should be at least 4 times the deflection.
(2) The ring height should be 15 to 20% of the outside diameter.
(3) The outside diameter should be as large as space permits.
(4) Normal ring taper is approximately 14° (θ)
(5) Allowable stresses are usually 160,000 psi for steel not machined after heat treatment and 200,000 psi for those machined after heat treatment. The ring spring can continue to function even though several of the rings are broken.

The lowest capacity commercial ring spring is in the order of 2 tons.

Another variation of the ring spring is the split ring spring in which each ring is slit on one side. The spring rate is greatly decreased, while the frictional action is retained. The contour of the spring is modified, so that the ring thickness is maximum 180° from the slit and tapers toward each side. The spring package is then arranged with alternating slit and thick sides.

Stranded Wire Springs

When an axial load is applied at the ends of a stranded helical spring, the material forming the helix is subject to a twisting moment. In this respect the stranded spring is not essentially different than a conventional spring (helical) made from a single homogeneous wire element. The outstanding characteristic of the stranded spring is an inherent tendency for the damping of high-velocity displacement of its coils, a characteristic not shared with the conventional spring. The damping in the stranded spring is due to a binding action existing between the twisted wires in consequence of the twisting moment acting on the strand.

In the stranded spring it is essential that the helix of the strand be opposite in direction to that of the coils of the spring. An applied load causes a twisting moment which tends to cause a wind-up of each helically formed wire. The wires of the strand are in contact with each other even in the unloaded state; and since there can be no appreciable wind-up, binding between the wires results. However, if both the strand and the coils of the spring have the same turn of helix, the twisting moment tends to unwind the strand, in which case the binding action is lost and the spring deflects as though it had a subnormal value of shear modulus.

For the purpose of load-deflection computations the stranded wire spring may be resolved into as many partial springs acting in

parallel as there are wires in the strand. The rate of deflection R may then be determined by the following formula:

$$R = Kn \frac{Gd^4}{8D^3 N}$$

K = a factor = 1.05 for 3 wire strands
n = number of wires in the strand
G = shear modulus = 11,500,000
d = diameter of wire in the strand
D = pitch diameter of spring coils
N = number of active coils

Stranded springs usually have one coil closed at ends but the ends cannot be closed as effectively as they can in conventional springs. For such unground end construction the number of active coils "N" can be estimated as the total number of coils less 1.2.

An accurate computation of the numerical value of the shear stress in the stranded spring is a complicated affair. However, if the spring is resolved into partial springs acting in parallel, as is assumed for load-deflection computations, an average value of the shear stress "S" may be obtained for each partial spring from the following formula:

$$S = \frac{Gdf}{D^2 N}$$

G = shear modulus
d = diameter of wire in the strand
f = deflection of the spring
D = pitch diameter of spring coils
N = number of active coils

For a given application in which the spring coils have a high velocity of displacement, springs which have relatively low values of statically computed stress at solid compression fail earlier than springs in which the stress at solid compression is relatively high. The performance of the gun driving springs apparently cannot be predicted by stress computations based on static assumptions. Driving springs should be designed to have the lowest possible mass.

Neglecting the effect of a difference in end coil construction and in the number of inactive coils, a 3-wire stranded spring, having wire diameters which are 68% of the diameter of the wire in a given conventional spring, will have substantially the same rate, pitch diameter, solid height and average computed stress as the conventional spring. Maximum fatigue life will result, when 3-wire strands are used, the strand being so proportioned that the ratio of the length along the strand axis in which a single wire makes one turn, to the strand diameter is between 5.0 and 5.5.

Two wire strands have a longer fatigue life than conventional round wire helical springs; but the results were very much inferior to those of 3-strand springs. Four or more wires are not stable unless a center wire is used.

The diameter which circumscribes the three round wires woven into a strand (and in contact with each other), will be 2.155 times the wire diameter; assuming the wires to lay on the corners of an isosceles triangle. Three wire strands having a ratio of twist in the strand to the strand diameter of 5. and 5.5, it is found that the strand diameter is very close to 2.18 times the wire diameter.

For either a conventional single-wire spring or a stranded spring in which the pitch angle of the coils is not substantially in excess of 10. degrees, a fair approximation of the time in milliseconds for the wave to travel from one end of the spring to the other can be obtained from the following formula:

$$T = .0354 \frac{D^2 N}{d}$$

D = pitch diameter of spring coils
N = number of active coils
d = diameter of wire

As the wave of displacement passes thru the coils of the spring immediately after firing, each element of the spring acquires the velocity of the free end upon the arrival of the wave at the element. The elements then tend to move at the acquired velocity until the motion is affected by the reflection of the wave from the fixed end of the spring. The initial reflection of the wave at the fixed end of the spring and the subsequent reflection from the free end are the causes of the extremely high dynamic stresses imparted to the spring.

According to transient wave theory the change of stress of an element of a spring varies directly as the change of velocity of the element. The damping effects of stranded springs become more effective as the velocity is increased. Stranded springs have little damping action at low velocities of displacement, and hence, compare most favorably with conventional springs when the displacement velocity is high.

It is believed that the greater damping action of stranded springs is effective in absorbing energy so that the reflected wave in a stranded spring has much lower energy content than the wave in a conventional spring. The substantial decrease in the energy content of the reflected waves decreases the dynamic displacement of the coils of the spring, proportionately reducing the stress in the spring.

The stress range is a more critical factor in the fatigue life of springs than the stress level, or the mean stress in the operating cycle.

Music wire is currently the most satisfactory stranded spring material. The fatigue life of pretempered chrome-silicon wire was distinctly inferior to that of music wire. The relatively low yield point of music wire enables it to yield and adjust itself to the strains incident to the stranding and coiling operations in making a stranded spring.

So long as the ratio of the pitch diameter of the coils to the diameter of the wires in the strand is not substantially smaller than approx. (13.) thirteen, stranded springs can be coiled automatically on standard spring coiling equipment.

As in the case of contentional music wire springs, it is essential that the stranded springs be stress relieved after coiling-heated to approx. 450°F for a minimum of 30 minutes. It is also essential to coil the stranded springs somewhat longer than finished length, and to remove the excess length of pressing the springs from free to solid height a sufficient number of times to assure that subsequent compression to solid will produce no further reduction in spring length. This pressing operation produces a beneficial residual stress pattern across the wire section in the unloaded spring; and as the operation promotes a minimum of set during the subsequent operation of the springs, it is just as vital where stranded springs are concerned as it is for conventional single wire springs.

The wave motion of the coils of a driving spring generates stresses which may exceed the stresses produced by static compression to solid height. If so, the spring will take additional set during the gun firing, especially during the first few rounds.

Shot peening increases the life of conventional single wire springs by up to 60%. Shot peening is not effective in increasing the fatigue life of stranded drive springs.

The Neg'ator Spring

This is a flat strip of coiled metal that has a nearly constant force level, at times even decreasing in force with deflection. A constant force had previously been achieved with dead weights or intricate cam or lever systems. Neg'ator is a trade name patented by the Hunter Spring Co. of Lansdale, Pa. that fabricates this item.

Principal features of this spring are:

1. Flat force - deflection curves

2. Extremely long deflections, or extensions, up to 50 times the length of the original spring

3. Ability to act without losses around corners

The important consideration in attaching a bayonet to the muzzle is the effect upon accuracy, as well as strength. Accuracy firings with and without a bayonet attached show a definite change in the center of impact, due to the eccentric mass. Close fits are of added importance. The bayonet latch should be concealed, so that it cannot be inadvertantly released when sparring. Barrel strength should be adequate to sustain severe thrust and chop loads via the bayonet.

While on the subject of muzzle and barrel, a discussion of barrel length will be followed by muzzle devices, such as muzzle boosters, flash hiders, silencers, and of most interest, muzzle brakes.

Barrel Length

The longer the barrel is in proportion to the bullet diameter, the less powder will burn at the muzzle. The .256mm Jap rifle with 31.4 inch barrel offers practically no flash and little smoke because powder burning is essentially completed before the bullet reaches the muzzle. Tests conducted of a 36 inch cal..30 barrel (M2 ball ammunition) show absolutely no flash and little smoke. Of course, a hot gun increases flash, and not even a 40 inch barrel was effective in eliminating flash, when firing long bursts.

Barrel lengths do not vary widely among commercial as well as military models. Theyusually conform to an original barrel length used in standardizing the cartridge. Of course, barrel length in handguns vary widely, and this is due to the intended use of the gun. Target models require a long sight radius, police models require lightness, and concealed weapons shortness. Velocities are not significantly affected, a 6 inch cal..38 barrel being 3 to 5% higher in velocity than a 4 inch barrel, and 10% higher than a 2" barrel.

For rifles, the loss in velocity (for several inches) is quite small, and can be overshadowed by other variables, such as tolerance in land and groove, powder temperature, powder measure, etc. The only reliable method of measuring effect of barrel length on muzzle velocity is to take one barrel, and using carefully measured loads of the same lot of ammunition, measure the velocity of a sample (at least 32 rounds) then cut the barrel off an inch (for example) at a time. This data should also confirm the interior ballisitcs theories formulated for that cartridge.

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Principal features of this spring are:

1. Flat force - deflection curves

2. Extremely long deflections, or extensions, up to 50 times the length of the original spring

3. Ability to act without losses around corners

4. High initial force

5. Ability to store and deliver twice as much energy as a common coil spring of the same initial volume.

The flat strip of metal is prestressed so that it possesses a strong natural curvature, and force must be exerted to straighten it. The spring gradient may be varied by the amount of prestress applied to each section.

The variables that affect spring force are:

1. Modulus of elasticity

2. Thickness of strip material

3. Width of strip material

4. Natural radius of curvature

This type of spring was tried as a magazine follower spring, for rifles, and as a drive spring for the T148 and T148E1 semi-automatic 3-shot 40mm grenade launchers.

Muzzle Devices

In the firing of a typical small arms cartridge, up to 42% of the potential energy of the propellant is exhausted at the muzzle. The topic of this subject will be on methods and devices utilized to either take advantage of this energy or to control it.

Items of mechanical hardware, such as front sights, bipods, bayonets, grenade launchers, etc. are routine in nature and need not be covered in detail. Grenade launchers are usually powered by blank cartridges, and the grenade weight causes the gas pressure level in the bore to be maintained at a higher level for a much longer period of time, therefore, for gas operated automatic weapons, the power delivered to the operating rod is substantially higher. In the M14 rifle this is controlled by a closure valve that blocks the gas orifice when grenades are to be launched. In the M1 rifle, the grenade launcher body includes a solid pin that opens a valve on the gas cylinder plug, thereby automatically venting some of the gases. Recoil impulse to the weapon structure, particularly the stock, barrel, and receiver is also high.

Conversely, for firing blank cartridges (without launching grenades or other missiles) the bore pressure is exhausted all too quickly to power the automatic weapon, so a "blank firing attachment" is added to the muzzle. In its simplest form, this is a muzzle cap with an orifice that restricts the outflow of gases. Blank firing attachments are usually colored bright red with a flag section visible along the sight line for safety purposes. It must be removed when firing ball ammunition, or the weapon will be severely damaged.

The important consideration in attaching a bayonet to the muzzle is the effect upon accuracy, as well as strength. Accuracy firings with and without a bayonet attached show a definite change in the center of impact, due to the eccentric mass. Close fits are of added importance. The bayonet latch should be concealed, so that it cannot be inadvertently released when sparring. Barrel strength should be adequate to sustain severe thrust and chop loads via the bayonet.

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A typical table of muzzle velocities versus length of barrel is given as follows:

Barrel Length	.270 Win. 150 gr.	.30-06 180 gr.	.300 Savage 150 gr.
24	2800	2700	2670
23	2770	2690	2655
22	2740	2675	2640
21	2705	2660	2620
20	2670	2640	2600
19	2635	2620	2575
18	2595	2590	2550
17	2550	2560	2570
16	2505	2525	2490

Shotguns, 12, 16, and 20 gauge were found to lose velocity linearly by 7 feet per second per inch, from a length of 27 in. to 20 in.

While we are on the subject of effect of barrel length on muzzle velocity, it would be well to discuss other factors that affect muzzle velocity. One particular facet is the constant demand for higher performance, in this case, a higher velocity.

Consider the question, "What is the limiting velocity obtainable in the small arms class?" Of course, this means increasing the charge/mass ratio to unusually high levels. Theoretically the absolute upper limit is approx. 13,000 feet per second. However, a number of penalties would have to be paid to achieve this. (For nitro-cellulose powder, it is approx. 9100 fps.)

An experiment was conducted to determine the effect of changing the charge/mass ratio upon muzzle velocity. To assure near-instantaneous and complete powder burning the grains were ground to a fine dust. (This is normally a dangerous practice). Pressures were in the order of 100,000 psi, and for high c/m ratios, extremely small bullets were used.

Accordingly, the following table was compiled:

(Note how the ratio of powder weight increase compares with the rate of muzzle velocity increase.)

c/m Ratio	Muzzle velocity (upper limit)
.2	2600 fps
.3	3000 fps
.8	4200 fps
3.2	6400 fps
5.8	7400 fps
11.0	8000 fps
22.0	9000 fps
44.0	9200 fps

Then, taking a typical commercial cartridge to correlate powder burning rate with c/m ratio vs. m.v.,

Powder	c/m RATIO	m.v.
Fast burning	.295	2800
Med. burning	.307	2800
Slow burning	.347	2800

As a practical example of extreme velocity effects, a classic example is the so-called Paris Gun of World War I. The Germans were bogged down on the Hindenburg line 75 miles from Paris, and developed three samples for the purpose of spreading panic among the civilians. These were reconstructed 15 inch 45 caliber artillery pieces tubed down to 8.26 inches. It is reported that the projectiles were waist high and the powder bags per shot twice as high as a man. The gun was set at 50°, fired at a muzzle velocity over 5200 fps and went about 12 miles up, so that approx. 3/4 of the trajectory was in near-vacuum.

One gun blew up, and the other two soon wore out, to be re-barrelled. A total of 200 rounds were fired.

Summarising the three methods of achieving high velocity:

(1) Use of light projectile: This is inefficient due to the extremely poor ballistic coefficient.

(2) Use of long gun with large powder charge: The barrel tube is too long to be practical, with serious barrel erosion, high pressures and recoil.

The ratio of bullet mass to gun volume must be small, and reviewing the interior ballistics theory that the muzzle energy is equal to the work done on the projectile,

$$(P\ avg) \times AL = 1/2\ m\ V^2$$

A = bore area
L = barrel length
P avg. = mean bore pressure

$$\therefore \frac{2\ P}{m/V_o} = V^2$$

The smaller the m/V_o ratio, the higher the velocity.
V_o = Bore volume

(3) This brings us to the third method of achieving a high velocity, that of a small caliber projectile in a large caliber gun, made possible by using a sabot of lightweight material.

One method of removing the sabot at the muzzle is by using a muzzle attachment called a "stripper". The bore is smooth, but the stripper is a short length of rifling, that imparts a spin to the segmented sabot, centrifugal force causing it to break apart evenly. Alloys of exotic metals are required for strippers of any endurance. The stripper length is approx. 4 to 6 calibers long.

Muzzle Booster

Muzzle boosters are used in recoil operated weapons to augment the energy delivered to the recoiling mass. For a conventional recoil cycle, the recoil mass is accelerated rearward while the projectile is accelerated forward. If the recoiling mass is too heavy, it will not have enough energy to complete the cycle satisfactorily for all firing conditions, particularly in adverse conditions and in extended firing schedules.

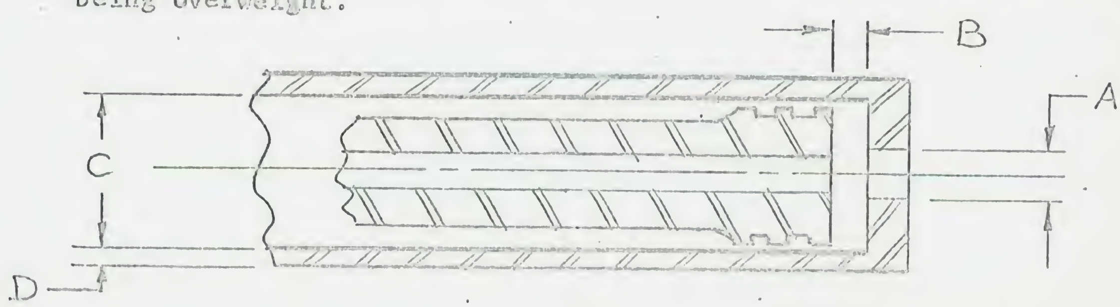
Of course, the weight is necessitated for strength and "heat sump" purposes, so a muzzle booster is incorporated into the barrel jacket which is fixed to the weapon receiver or frame, and does not recoil with the barrel. It traps muzzle gases just as the projectile exits, so that the gases impinge between the capped end of the barrel jacket (muzzle booster) and the barrel muzzle face, thus adding to the recoil impulse. In this way, some of the excess energy that normally escapes at the muzzle is trapped and put to work. Critical dimensions include the following:

A. Exit bore diameter, which must not interfere with maximum bullet yaw angle, nor be too large.

B. Inboard length, which must compensate for increase in barrel length due to thermal expansion in prolonged firing.

C. Barrel bearing diameter, which should not be too small, thus binding when barrel muzzle expands thermally, nor should it be large enough to permit excessive barrel vs. jacket mis-alignment during muzzle vibration, causing interference of bullet with dia. "A".

D. Thickness of jacket, to provide tensile strength sufficient to resist strain caused by gas pressure acting on booster cap, without being overweight.



Flash Suppression

Muzzle flash is caused by excessive temperature and pressure of muzzle gases. A number of chemical additives have been added to the propellant to reduce the tendency to flash. Most have been salts of alkali metals. The most effective was cesium iodide, but the most commonly used are salts of potassium, or potassium sulfate.

Mechanical flash suppressors are most effective. A cone type flash hider was first used, but is limited in effective flash suppressive action. The average cone type flash hider has a 12 to 18^{o} included angle, and a length of 3 to 6 inches. (7.62mm)

The bar type flash suppressor is most effective in the manner that the gases expand through the slots, breaking up the continuity of the flow, thus preventing shock formation. An odd number of bars is usually necessary, and the end of the slot nearest the muzzle is at right angles, because gas increases in velocity when it turns at right angles, externally, mixing quicker with the air, for a cooling effect.

The width of the bar is slightly wider (approx. .02 inch) than the slot opposite it, on symmetrical designs. For rifles, the lower bar is usually wider, in order that it may function as compensator in reducing muzzle climb.

The slot area facing the bore should be at least 20 times the bore area. The suppressor is usually open-ended, except on ground weapons, in which a closed end is used to prevent brush-spearing discomfort.

Silencers

The use of silencers is usually limited to low powered weapons. A silenced weapon has advantages on certain applications where firing is necessary without revealing one's position.

The principle of a silencer is similar to that of an automobile muffler, in which the energy of the gases is reduced by an expansion chamber and baffle system. Silencers are usually cylindrical in shape and project in front of the barrel as well as around it. The weight and bulk increases as a function of the degree of noise suppression. That is, there may be a compromise between silencing and volume.

Only projectiles traveling at sub-sonic velocities can be completely silenced, since a projectile at supersonic velocity sets up a shock wave in the atmosphere, creating a sharp cracking sound along the trajectory until the velocity drops to sub-sonic.

The weapon discharge, however can be silenced, and, of course, the silencer is heavy, since it must withstand the muzzle pressure.

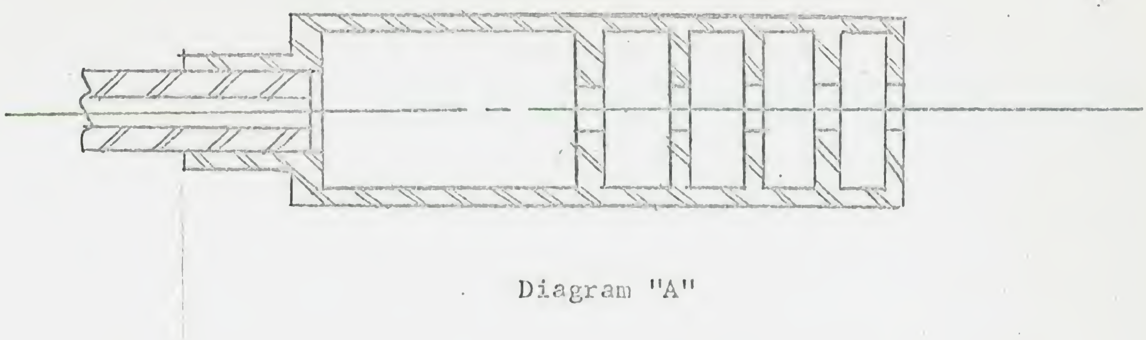


Diagram "A"

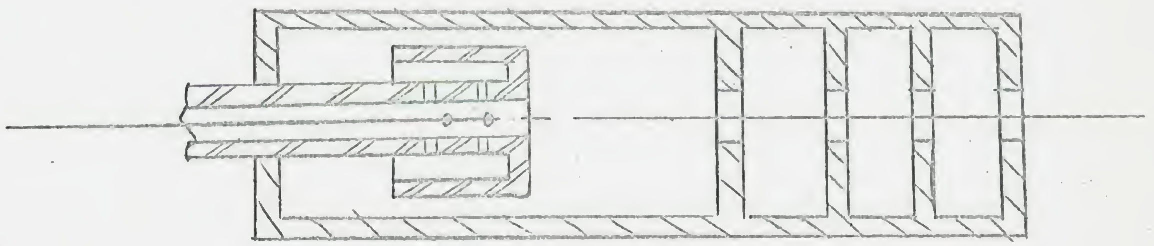


Diagram "B"

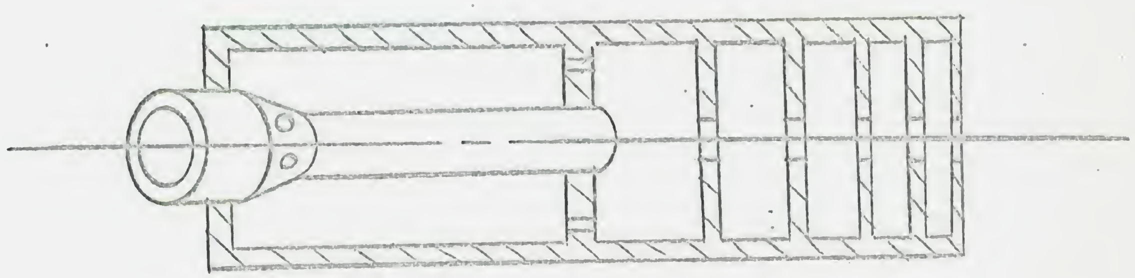


Diagram "C"

TYPICAL SILENCER DESIGNS

Silencers are usually fitted to weapons with fixed barrels. This is because the weight of a recoiling primary mass with/without the silencer would vary too greatly. In this case, the silencer could be mounted on a non-recoiling barrel jacket.

The common revolver cannot be completely silenced due to the gas leakage at the barrel/cylinder interface.

One typical silencer is shown in diagram A.

An initial chamber allows muzzle blast to be reduced and the following smaller ones dissipate gas energy, so that noise is reduced. The bullet hole should be as small as possible commensurate with bullet yaw. In some cases a self-sealing rubber pad is inserted, but this eventually burns, and is limited to a few rounds before replacement, if desired.

The muffler, or silencer, will act as a muzzle brake, become overheated in extended firing, and requires frequent cleaning.

In diagram B the sharp muzzle blast peak is reduced, being vented about an additional baffle. Muzzle velocity is not appreciably changed.

In order to reduce velocity of standard supersonic rounds (to avoid special ammo) some of the gas may be exhausted into an expansion chamber. For example, in diagram C the 9mm Parabellum cartridge (m.v.1330 fps) could employ this system for optimum silencing.

By law, silencers are not permitted on commercial weapons, but certain types of "sound moderators" may be available.

The noise level attained can be measured by a microphone connected to a cathode ray oscillograph, with photographs taken of the sound waves.

Muzzle Brake

A muzzle brake functions to reduce recoil effects by trapping gas at the muzzle and causing a forward impulse to act on the weapon.

Usually a series of baffles are formed in a muzzle brake so that the gases pass through and are diverted to the rear.

The gases should not be completely reversed or a blast wave may reach the shooter's face, eyes, or ears. The ports should be symmetrical, so that the gun is not turned, and should not be directed downward where they would raise a dust cloud. A muzzle brake can also be integrated with a compensator, to keep upward muzzle jump to a minimum.

Essentially, the action of the muzzle gases on a muzzle brake is the same as that of (for comparative and demonstrative purposes) a jet of high pressure steam on the turbine blades of a turbine. The kinematic analysis of each is similar. The gas pressure reduces quickly at the muzzle and the gases escape with a kinetic energy equal to the pressure difference only in the case in which a Laval expansion nozzle is fixed at the muzzle. The gases must pass with a high velocity from the muzzle to the blade (brake). The design of the nozzle must accordingly not impede this flow.

Outflow velocities in the case of simple parallel openings reveal that the critical velocity is far surpassed.

The cross-sectional boundary where the gases are not yet mixed with the air is shown in the accompanying sketch, in graphic scale. This represents the "jet border" within which the muzzle brake must act, also shown in part b of the sketch.

The time that the projectile travels from the muzzle to the brake vane, or blade, is the effective period of the muzzle brake. Gas mass is also significant, so the higher charge-to-mass ratio (c/m) loads are more efficient in muzzle brake action. That is, weapons using ammunition with low charge weights cannot effectively benefit from use of a muzzle brake.

To determine the efficiency of the muzzle brake, fire the weapon at 0° elevation without the muzzle brake, then with the muzzle brake.

As the difference in the lengths of the recoils is only a few millimeters, one can assume in practice that in both cases, with the maximum recoil velocity, the recoil lengths are the same, and that the maximum recoil velocity occurs at the end of the after effect. The brake force K_x may have a constant value.

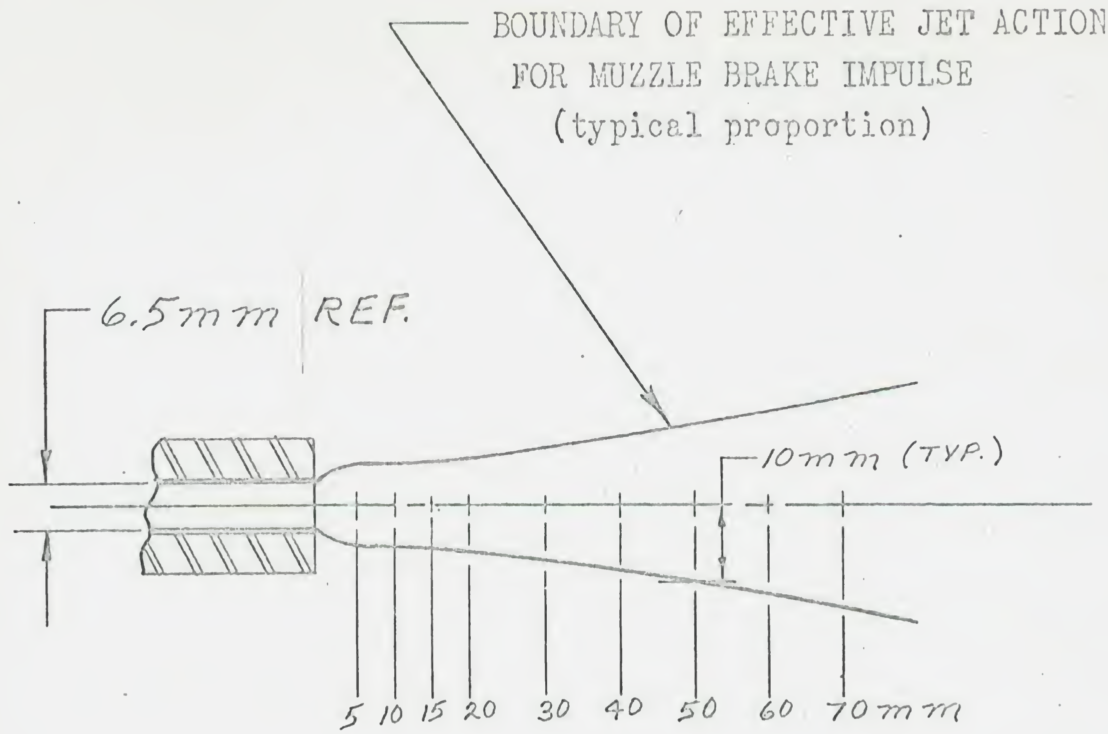
K_x = The constant brake force without muzzle brake,

K'_x = The constant brake force with muzzle brake,

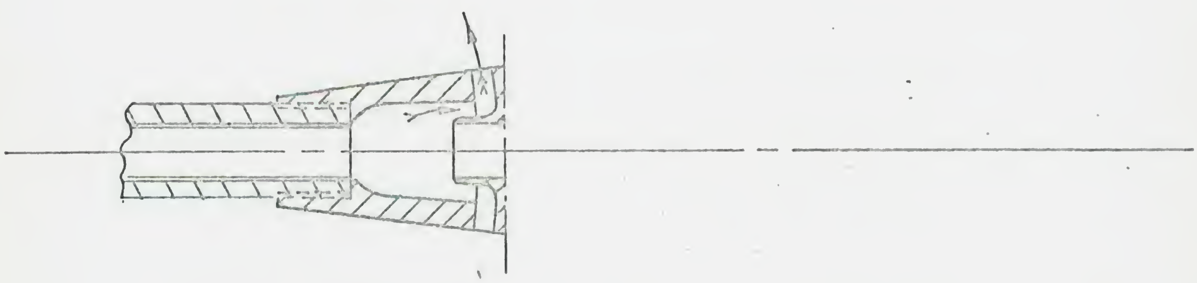
Gr = Weight of recoiling parts.

I. Without muzzle brake: $K_x \cdot S3 = \frac{Gr}{2 \cdot g} \cdot V^2 \max = E$,

$$K_x = \frac{2 \cdot g \cdot S3}{Gr} \cdot V^2 \max \tag{1}$$



RESULTANT DESIGN OF MUZZLE BRAKE ELEMENT



II. With muzzle brake: $K'x \cdot s_3 = \frac{G'r}{2.g} \cdot V'^2 \max = E,$

(2) $$K'_x = \frac{G'r}{2.g.s_3} \cdot V'^2 \max$$

$V' \max \leq V \max$ and in both cases the lengths of the recoils after the after effect are equal because the openings through which the flow takes place are closed only after the end of the recoil. From this it follows that:

(3) $$K'_x < K_x.$$

and the efficiency of the muzzle brake is:

(4) $$y = \frac{E - E'}{E} \cdot (100)$$

Since G_r is almost euqual to G'_r it follows that:

(5) $$y = \left[1 - \frac{V'^2 \max}{V^2 \max}\right] \cdot 100$$

If, also, $s_2 = s'_2$ we may write:

(6) $$E - E' = \frac{Px \cdot s_2}{2}$$

S_1 = Length of recoil, projectile in barrel
S_2 = Length of recoil during the after effect
S_3 = Length of recoil after the after effect

Consequently, the force of the muzzle brake is:

(7) $$Px = \frac{2 \cdot (E - E')}{s_2}$$

Here it was also taken for granted that

$s_2 = s'_2$ and $K_x = K'_x$ (until the end of the after effect)

The Reduction of the Recoil

Without the muzzle brake the total recoil length is:

(8) $$S = s_1 + s_2 + s_3$$

The brake force on the path s_3 is: K_{x3} = Constant

With the muzzle brake the total recoil length is:

(9) $$S' = s'_1 + s'_2 + s'_3$$

The brake force on the path s'_3 is: K'_{x3} = Constant

(10) and $$K_{x3} > K'_{x3} \text{ , besides } s_3 \sim s'_3$$

The question now is how great will be the length of recoil ($s''3$) when a muzzle brake is used, if the brake force K_{x3} is the same as without a muzzle brake. We must have:

(11) $$K_{x3} \cdot s''_3 = K'x3 \cdot s'_3 \; ; \quad s''_3 = \frac{K'x3 \cdot s'3}{Kx3}$$

And the reduced length of recoil is: $S_r = s'_1 + s'_2 + s''_3 = s_1 + s_2 + s''_3$ because up until the cessation of the after effect, the recoil lengths are almost the same with and without the muzzle brake. The amount of the shortened recoil is:

(12) $$m = \frac{S - (s'_1 + s'_2 + s''_3)}{S} = \frac{(s_1 + s_2 + s_3) - (s'_1 + s'_3 + s''_3)}{s_1 + s_2 + s_3} =$$

$$= \frac{s_3 - s''_3}{S}$$

Since:

(13) $$s_1 \sim s'_1 ; \; s_2 \sim S'_2$$

VI Weapon Analysis

This is the most critical, constructive, and informative phase of the weapon engineering cycle. Here the designer learns to correct his mistakes, adjust his theories, redesign his mechanisms, and join with others in evaluating performance. The instrumented test vehicle provides time and motion data so that velocites of moving parts are determined, and energy dissipation compared with the weapon cycle.

In this section, the time-displacement camera and resultant capabilities are demonstrated followed by an actual analysis of a recoil operated machine gun under development. Finally, a visual analysis of the Soviet AK-47 assault rifle is given. Firing data and time-displacement records were not available, but the description will reveal a number of features in this design that should be of interest to all small arms design engineers.

Time-Displacement Data

The time displacement record is one of the most useful instruments that a weapon designer can use in the development of an automatic weapons. A typical drum-type T-D camera is shown in the accompanying sketch. With it, the following is determined:

(a.) Motion of selected component of weapon in distance traversed

(b.) Time of component traverse

(c.) Dwell time between shots

(d.) Rate of component traverse during cycle of operations

(e.) Component bounce, if any

(f.) Deflection of mount, or semi-rigid receiver, during firing

(g.) Reproducibility of component travel in a burst of fire

(h.) Rate of fire

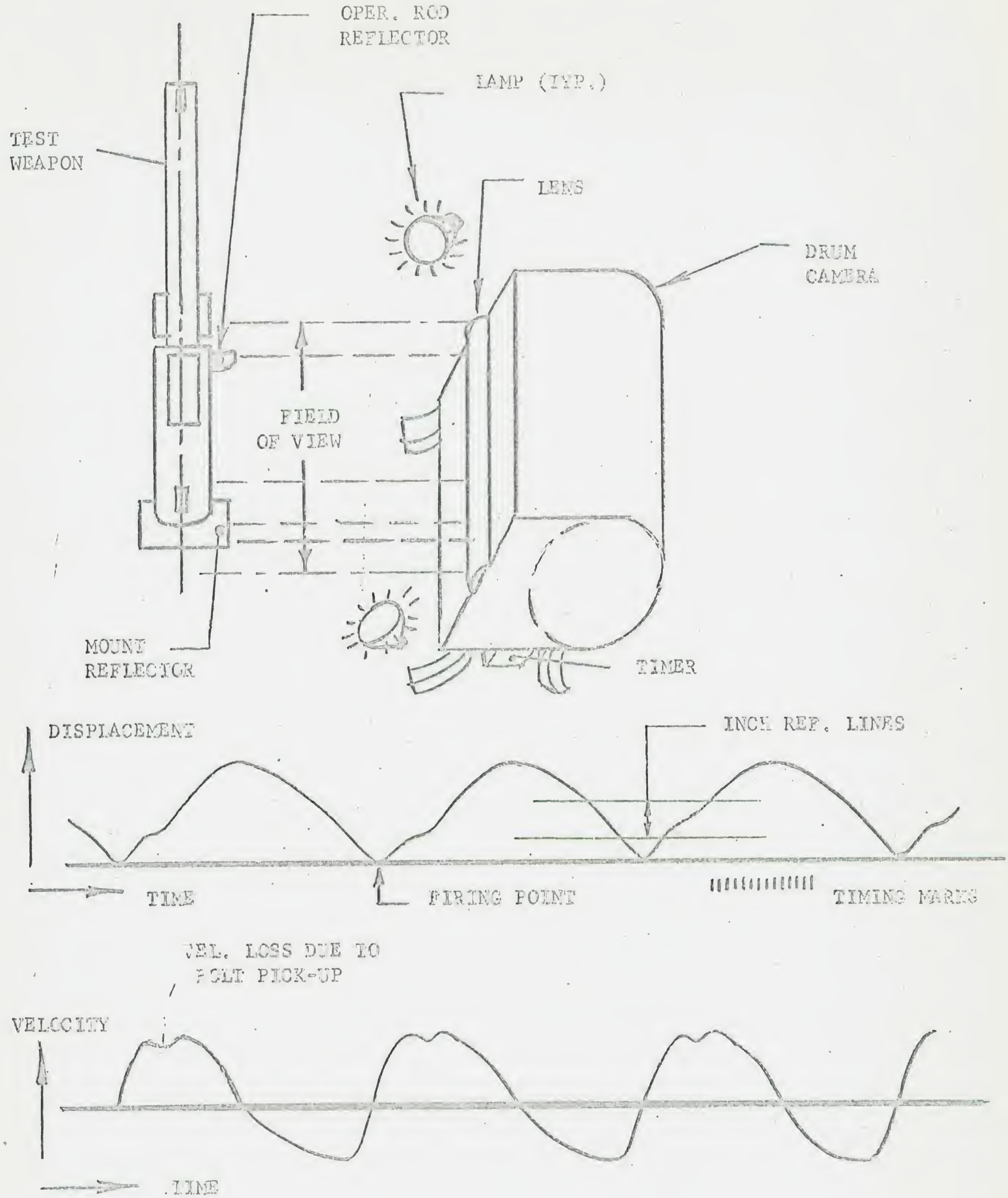
(i.) Time required for mount motion to dampen out.

From the time-displacement curve, the following is derived:

a.) Absolute motion of given component independent of mount deflection,

b.) Velocity of a given component through out its traverse,

TEST WEAPON TIME-DISPLACEMENT DATA



TYPICAL M60 M.G. T-D & T-V DATA - BURST FIRING

c.) Acceleration and/or deceleration of the same component,

d.) Energy balance during recoil and counter-recoil motion,

e.) Coefficient of restitution of buffer mechanism.

$$= \frac{Er - Ec'r}{Er}$$

In a typical test, the weapon is mounted in a cradle, or test fixture, and a reflector pin attached to the desired component, say operating rod, and another reference pin mounted on the receiver, or mount. The T-D camera is positioned perpendicular to the weapon, in a brace of spotlights, so that the reference pins reflect light onto the camera's lens.

Within the camera is a 12" dia. rotary drum that will be revolving at a high constant speed at the time of firing. A series of timing marks (indicating milliseconds) will be printed on the roll to indicate the exact speed. A test exposure is made to determine the linear scale, called "inch marks".

On firing, the light reflected from the reference pin will automatically print a curve of operating rod motion with respect to camera drum rotation (time) and a companion curve of mount deflection.

After development of the print, the T-D curve will be a permanent record of the operating rod motion diagram. A brief visual inspection will immediately determine whether or not operating rod motion was smooth and efficient, or "jerky". By viewing with the eye tangent to the curve, discontinuties in motion are readily apparent. Several single shots or a burst of fire may be recorded on a single sheet, so that the slope of the T-D curve can be readily compared for all rounds as well as time span to recoil and counter-recoil motion.

Sudden shifts in slope indicate change in velocity, energy loss, or work done. This is exemplified in the diagram showing recoil motion and attendant velocity diagram of the 7.62mm M60 machine gun. Note how the velocity drops suddenly when the operating rod picks up the relatively heavy bolt and does work in extracting the cartridge case. Then a velocity increase indicates effect of residual chamber pressure (blowback action).

The following data should be marked on one corner of the T-D record for reference:

A) Date of test

B) Weapon model and serial number

C) Number or rounds fired

D) Ammunition lot number

E) Weight of recoiling parts or other components of interest

F) Purpose of test or J.O. #

G) Curve Number

Before evaluating the "operating rod" motion curve, remember that the indicated curve is a result of operating rod motion plus or minus mount motion, depending upon the mount motion diagram. Therefore, a "corrective" operating rod motion diagram must be constructed. This is done as follows:

a) Layout a series of vertical evenly-spaced increments, e.g. .1 inch apart, so that they pass through both the mount motion diagram and the operating rod motion diagram.

b) When both curves are in recoil, subtract the mount motion at each increment from the operating rod motion.

c) When the mount has gone forward past the starting plane then add the mount motion at each increment to the operating rod motion.

The resultant series of points will form the corrected curve of operating rod motion with respect to the receiver. The slope of this curve at any point is the velocity at that point. That is, the tangent of the angle, when correct for time and linear scales, is the operating rod velocity, as follows:

a.) To determine the time scale, measure the length of 20 msec. (21 lines) and divide by 20 (Never use a single spacing)

b.) To determine the linear scale, measure the distance between "inch marks".

c.) Remembering that velocity equals "distance" divided by "time", then:

$$V = \frac{d}{t} = \frac{1/\text{ linear scale} \times 1/12 \tan \theta}{1/\text{time scale}}$$

For example, in a T-D curve where the "inch marks" are 2.03 inches apart, and the "timing marks" are .261 inches apart,

$$V = \frac{1/2.03 \times 1/12 \tan \theta}{1/.261 \times 10^{-3}} = 10.7 \tan \theta$$

$\tan \theta$ = tangent of the curve at any selected point.

d.) Therefore, to obtain a velocity diagram, measure the slope at each increment, and multiply the tangent of the angle by the constant (in this example, 10.7)

To restate the example, calculate the velocity for a slope in which the angle is 45°. The tangent of 45° is 1.0, therefore, this will be the constant that corrects for linear and time scales.

Velocities at any other slope is then merely the tangent of the angle times the velocity calculated for a 45° slope.

If a time-displacement record shrinks or stretches in time, the scale will also move accordingly. Therefore, the record will remain accurate.

An acceleration curve may also be obtained for the operating rod motion as follows:

a.) After laying out the velocity curve, to some suitable scale, say 1" = 10 feet/second and smoothly connecting the points plotted, the slope of the velocity curve is an indication of the acceleration.

An increasing velocity curve shows positive acceleration, while a decreasing velocity indicates deceleration, with attendant forces.

b.) Since $a = v/t$,

$$a = \tan \theta' \times \text{velocity scale} / 1/ \text{time scale} \times 10^{-3}$$

As in the previous example, where in the velocity scale, 1" = 10 fps and in the time scale, 1 msec. = .261 in. This could also be stated as 1" = 3.83 ms, so that the inverse time function would be used in the above formula. $a = \dfrac{\tan \theta' \times 10}{1/.261 \times 10^{-3}}$

$\therefore a = 2610 \tan \theta'$

Therefore the tangent of the observed angle X 2610 = the acceleration at the point observed.

The firing rate is easily calculated by measuring the horizontal distance on the T-D curve from a point on one round to a similar point on the succeeding round and applying the time scale.

This may be done two ways:

a.) measured distance / in/msec. = time,

or b.) measured distance X msec./in = time,

depending upon how the time scale is factored.

In the example noted, "X"/.261 = cyclic time.

$$\text{Rate of fire} = \frac{60}{\text{cyclic time}} \text{ (in rpm)}$$

Working from layouts, the positions of key events should be marked on the T-D curves; notably the position where the operating rod picks up the bolt, or where the accelerator works, or where the feed system operates. Knowing the masses involved, then energy used for each function may be determined. This can be compared with energy required to do the work, such as feeding, etc.

As an example, the following data is taken from a study in recoil operated weapons described in Chinn's Vol. IV of "The Machine Gun" pg. 112. Here Time-Travel and Time-Velocity Curves of a Barrel - Accelerator - Bolt function are illustrated.

With the given velocities, Energy values are determined by the formula $E = \frac{W V^2}{2 g}$

Accordingly, the following was determined:

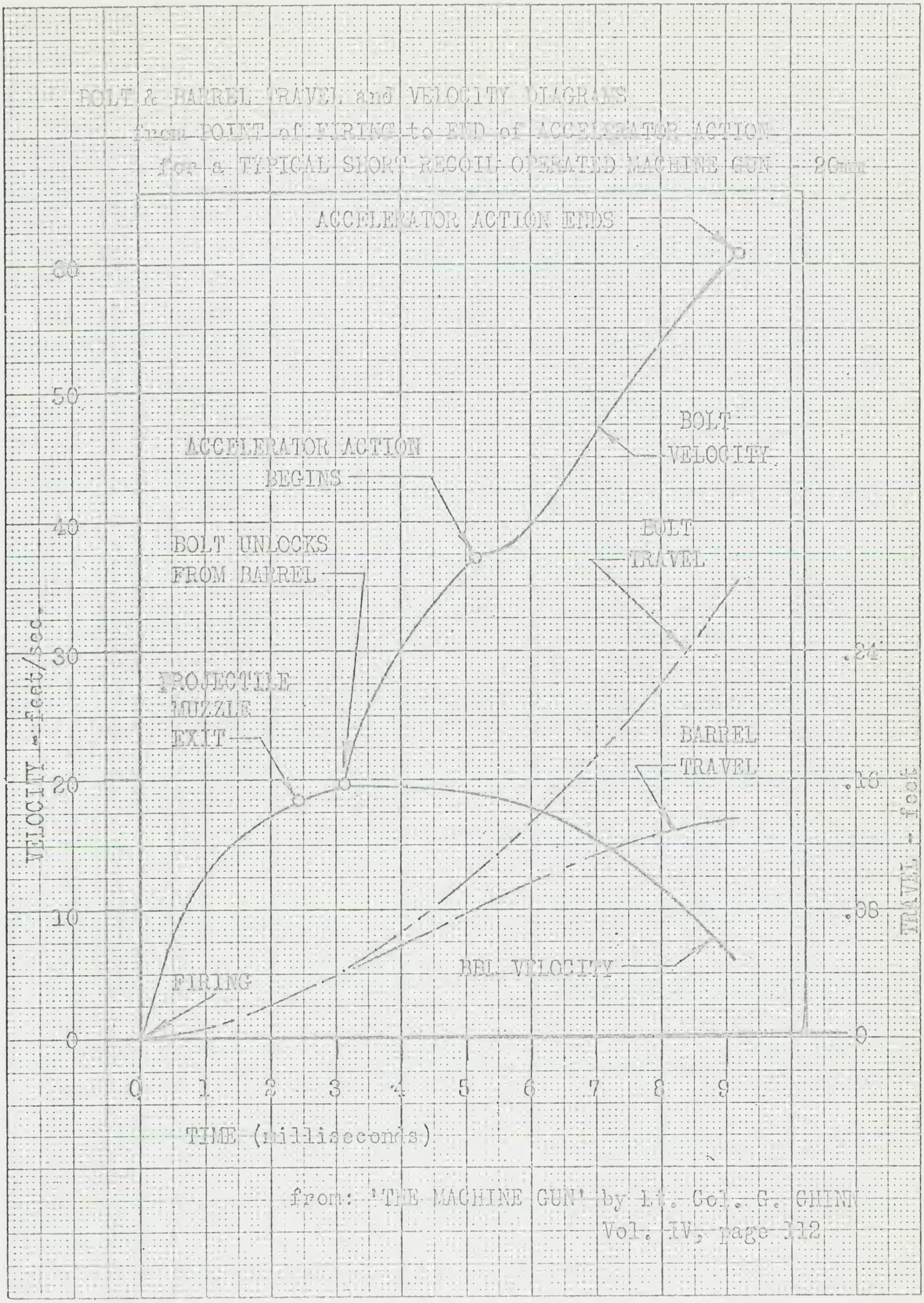
	Bolt, 5 lb.	Barrel, 45 lb.
At Unlocking	18 fps = 25.2 ft. - lb.	18 fps. = 226 ft. - lb.
Start of Acc.	38 fps = 112. ft. - lb.	17 fps. = 202 ft. - lb.
End of Acc.	60 fps = 280 ft. - lb.	7 fps. = 34 ft. - lb.

Thus, the bolt gained 168 ft. - lb. energy while the barrel lost 168 ft.-lb. during the period of acceleration.

Actually, there would be other losses that were not shown here, such as friction, etc.

The bolt energy gain between unlocking and start of acceleration is interesting, and is ascribed to the effect of blowback action.

Translating this theory to the chamber forces, we have the following relationship:


BOLT & BARREL TRAVEL and VELOCITY DIAGRAMS
from POINT of FIRING to END of ACCELERATOR ACTION
for a TYPICAL SHORT RECOIL OPERATED MACHINE GUN - 20mm
ACCELERATOR ACTION ENDS
BOLT VELOCITY
ACCELERATOR ACTION BEGINS
BOLT UNLOCKS FROM BARREL
BOLT TRAVEL
PROJECTILE MUZZLE EXIT
BARREL TRAVEL
BBL VELOCITY
FIRING
VELOCITY - feet/sec.
TRAVEL - feet
60
50
40
30
20
10
0
.24
.16
.08
0
0 1 2 3 4 5 6 7 8 9
TIME (milliseconds)
from: 'THE MACHINE GUN' by Lt. Col. G. CHINN
Vol. IV, page 112

Given an increase in bolt energy of from 25 to 112 ft. lb. = 87 ft.-lb.

$$I^2 = \frac{2\,E\,W}{g} = \frac{870}{32.2} = 26.4$$

$\therefore$ Added Impulse = 5.2 lb.-sec.

time = .002 sec.

Since $I = F\,t$, $F = 5.2/.002 = 2600$ lb.

For the 20mm cartridge, the case dia. (inside) = .9",
Area = .635 in.^{2} Residual pressure = F/A = 2600/.635 = 4100 psi. As an average pressure, this is reasonable, since the pressure at the muzzle exit is given as 5000 psi.

In summation, the energy calculated for the moving parts should be reasonably traced back to the interior ballistic data for either the chamber and/or gas cylinder.

Analysis of a Recoil-Operated Machine Gun

As a typical example of weapon analysis, consider a 40mm short recoil operated machine gun under development which, in recent firing tests at APG, displayed an undesirably high stoppage rate. This weapon is designed so that in its cycle of operations, all of the required motions are in series, and depend upon mechanical "signals" at the completion of one phase to initiate the following phase in the cycle. This principle was employed to optimize reliability, since the required rate of fire was low enough to facilitate this approach. That is, the weapon cycle would not require two separate mechanisms to function simultaneously. This is particularly desirable in the feed mechanism.

The analysis will correlate the cartridge impulse with the recoil and counter-recoil impulse of the bolt and barrel assembly This will determine whether the principal motive power is being efficiently utilized in operating the weapon. In studying a time displacement diagram, a study of bolt and barrel velocity and energy diagrams will reveal any inefficiencies in the distribution of energy during the weapon cycle.

The cartridge impulse is 13.1 lb.-sec. This is taken from the formula $I = m\,V$, where projectile weight is .532 lb., and velocity averages 790 fps. Gas impulse is negligible, since the powder charge is small, and a high/low pressure system is used.

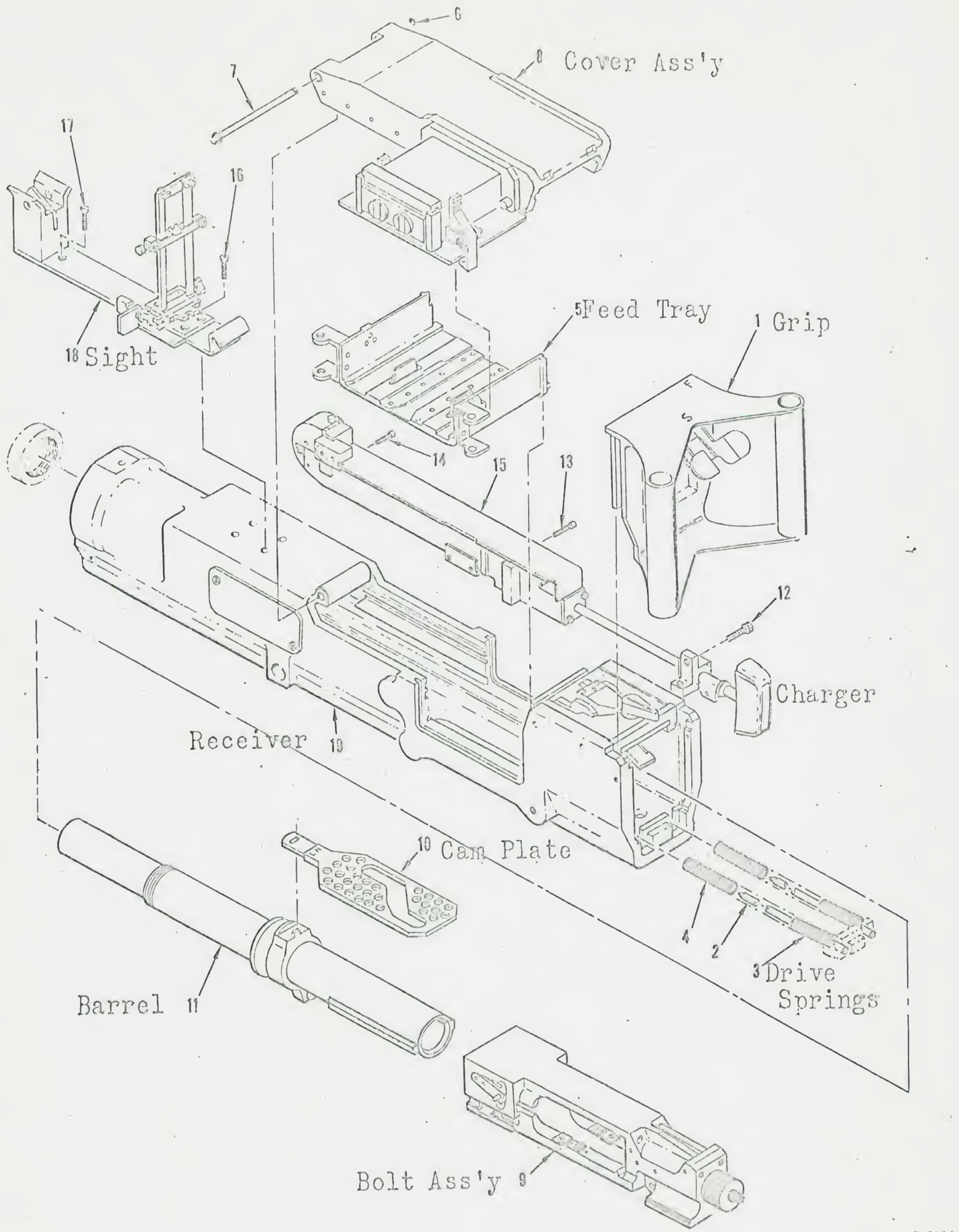


Figure 1-4. XM175 grenade launcher components

Physical data taken from the weapon is as follows:

Barrel assembly weight:	5.07 lb.
Feed cam	.60
Bolt assembly	6.90
Feed slide assembly	1.10

Combined bolt and barrel weight in counter-recoil (with linked round) is 13.50 lb., and in recoil (with linked case) is 13.03 lb.

Accordingly, the impulses were compared. Counter-recoil impulse is 2.9 lb.-sec. and recoil impulse is 10.3 lb.-sec. for a total of 13.2 lb.-sec. which is in agreement with the cartridge impulse of 13.1 lb.-sec. Velocities were determined from averaging a series of time displacement curves as shown. C'recoil velocity is 7.0 fps, while recoil velocity is 25.48 fps.

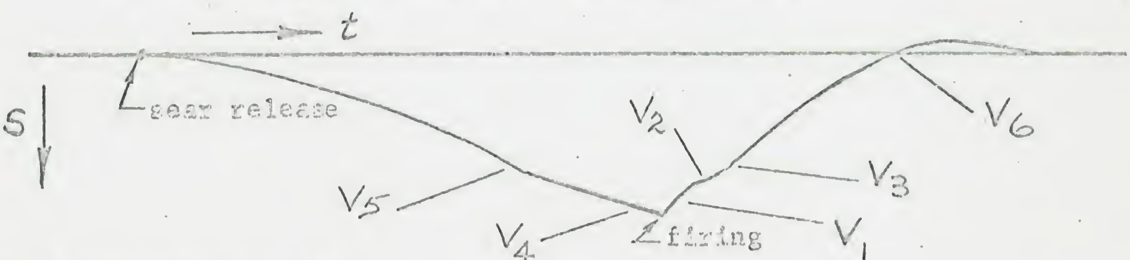
The data from these T-D curves was found to be quite regular from shot to shot, and is as follows: (V_1 to V_6 indicates velocities at principal locations in the cycle) (shown below)

One time displacement curve was analyzed in detail and segmented into 52 increments to show displacement, velocity, and remaining energy for the full bolt-barrel traverse.

The energy loss where the bolt picks up the barrel during counter-recoil traverse is obvious, and indicates inefficiency in the relationship between barrel unlatching and bolt locking.

Schematic of T-D Curve

The velocities and times noted are as follows:



V_1 = Maximum velocity
V_2 = Sudden velocity loss
V_3 = Resumption of velocity level
V_4 = Approaching firing
V_5 = Bolt approaching barrel pickup
V_6 = Bolt approaching sear line
t c'r = time from sear release to firing
t recoil = time from firing to bolt hook-up on sear
t feed = feed time, taken as 38 msec if not available on T-D
curve, so that a comparable cyclic rate can be determined.

Curve	V1	V2	V3	V4	V5	V6	t c'k	t sec	RATE
#4	25.3 fps	18.5 fps	21.8 fps	7.1 fps	10.1 fps	6.07 fps	83.6 ms	57.7 ms	335 rpm
#5	24.9	18.5	21.9	7.5	11.3	7.7	80	55	347
#6	was a 5 round burst as follows:								
6a	26.	18.7	22.6	6	7	8.1	105	50	339
6b	25.7	18.5	22.6	7.4	10.5	9.9	82	48	350
6c	25.7	19.	22.6	7.3	10.8	10.3	85	47	340
6d	25.7	18.7	22.4	7.2	10.1	10.7	85	46	360
6e	26	19	22.4	7.3	10.7	11.2	83	46	
#3	26.5	17.4	21.5	7.1	10.5	6.6	85	62	325
#2	25.1	17.6	21	6.7	indistinguishable				
#1	23.9			6.5	"				
Avg.	25.5	18.4	22.1	7.0	10.1	8.8			342

[illegible]: curve [illegible] is the sample detailed in diagram [illegible]

The sharp velocity and energy fluctuations at points 31 and 32 where the feed cam is shifting from a 28° slope to a 43-1/2° slope are most irregular.

The rapid energy decline at points 41 and 42 indicate binding of the bolt in the receiver during recoil.

Feed spring loads were taken as well as feed cocking cam measurements and the feed lever crank ratio. The feed springs have an assembled load of 16 lb. and a rate of 12 lb./inch for each of the two springs. The cam has a dual camming slope of 28° for 1/3 of its stroke, then a slope of 43-1/2° for the remaining 2/3 stroke. The feed lever has a cam arm of 2.2 inches and a feed slide arm of 4.3 inches for a ratio of 1.95.

During recoil, the feed mechanism is cocked by the above cam and arm. The feed slide is spring powered, and remains cocked until the bolt is fully opened. The bolt then signals the feed slide to begin feeding. The bolt remains opened until the feed slide completes its stroke, then the feed slide signals the bolt to begin its chambering and firing stroke (in counter-recoil).

Time-displacement records show that the bolt bounces after it signals the feed slide. Therefore, the cartridge is not properly fed and a stoppage occurs.

The coefficient of energy return by the bolt buffer is .60, and could be improved to .80 by use of a hydraulic buffer. The bolt bounce is also augmented by drive spring surge. This can be alleviated by the use of stranded wire drive springs.

A study of the feed cam design is necessary because of the unusual behavior of the barrel velocity curve, cam profile, and resultant feed slide velocity diagram.

The feed slide is subjected to an unusually high acceleration at the point where the cam changes slope. The velocity changes from 25.5 fps to 42.6 fps in .3 milliseconds. This results in an acceleration of 57,000 fps^2, so a high reaction load will occur.

F_1 = m a = 1.1X57,000/g = 1950 lb.
F_2 = 1950 X 1.95 (lever arm ratio) = <u>3800 lb.</u>

Accordingly a simple modification of the cam path, to eliminate the sudden change in slope is recommended.

This is a radius tangent to the initial 28° slope. While this is not optimum, it frees the feed mechanism of the high acceleration loads.

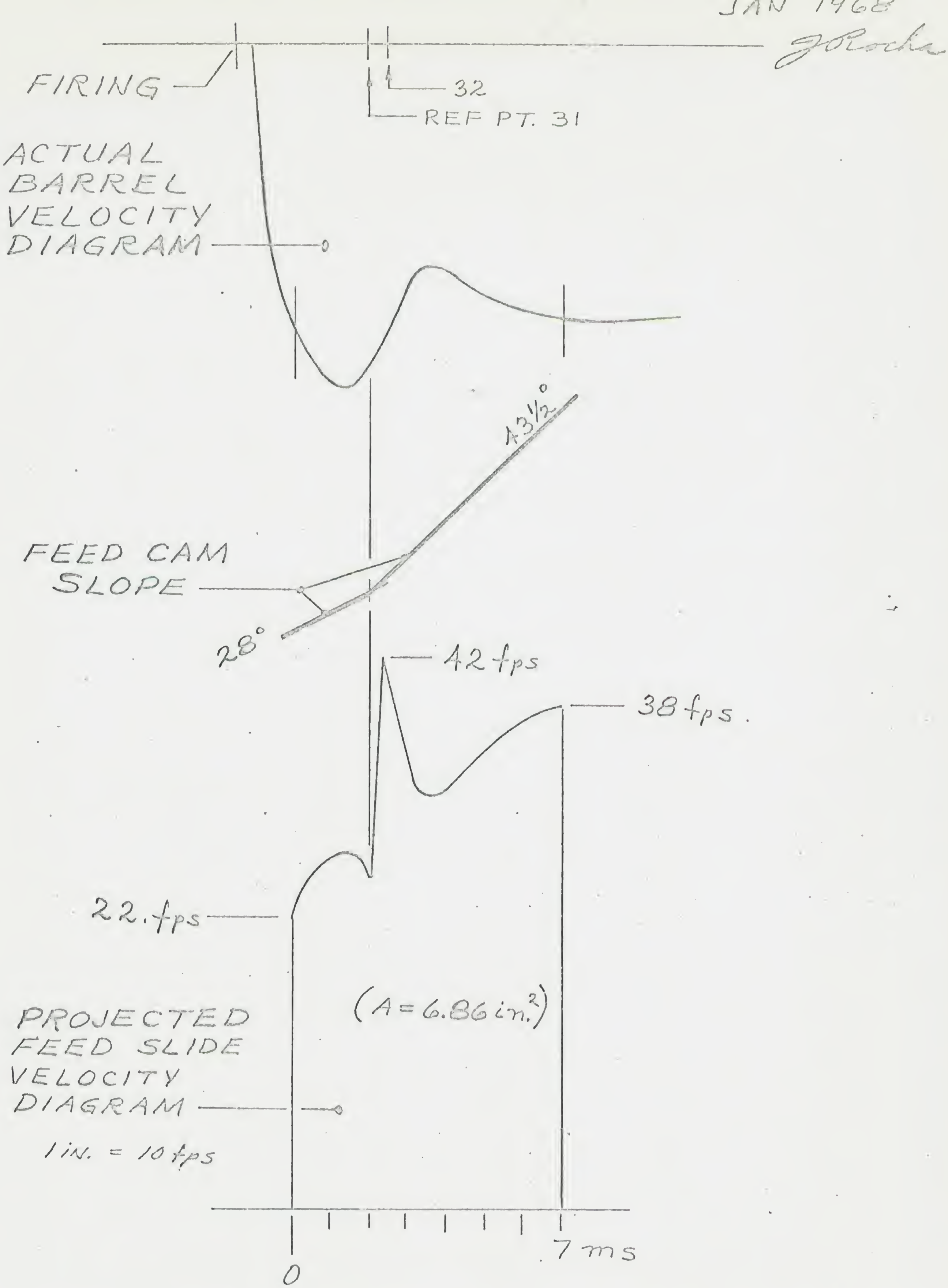
A complete redesign of the cam is in order, and a cycloidal cam is recommended, of the form:

displacement $Y = \frac{h}{\pi} (\emptyset - 1/2 \text{ SIN } 2\,\emptyset)$

The cycloidal cam has the lowest vibration, wear, stress, noise and shock. The reason for its excellent performance is that there is no sudden change in acceleration at the intersection of the dwell periods and the rise curve. It is also easy starting and the side thrust of the translating follower is low.

A time displacement diagram of actual feed slide motion during cocking was evaluated and the erratic velocity diagram indicates bouncing of the roller, and severe distortion of the feed slide arm. The peak velocity is in the order of 52-54 fps. The area under the curve is 6.81 in., while in the previous diagram the area is 6.86 inches. This indicates that the total energy under each curve is similar; therefore confirming the mathematical process.

In a further study of this feed system, it is noted that the feed pawls engage the cartridge at a relatively high position on the periphery. The angle of loading to the cartridge center is 26°30', so that the load components F_h = .89 F and F_v = .44F indicate a high vertical component tending to roll the cartridge.



DERIVATION of FEED CAM LOAD DATA-"A"

JAN. 68
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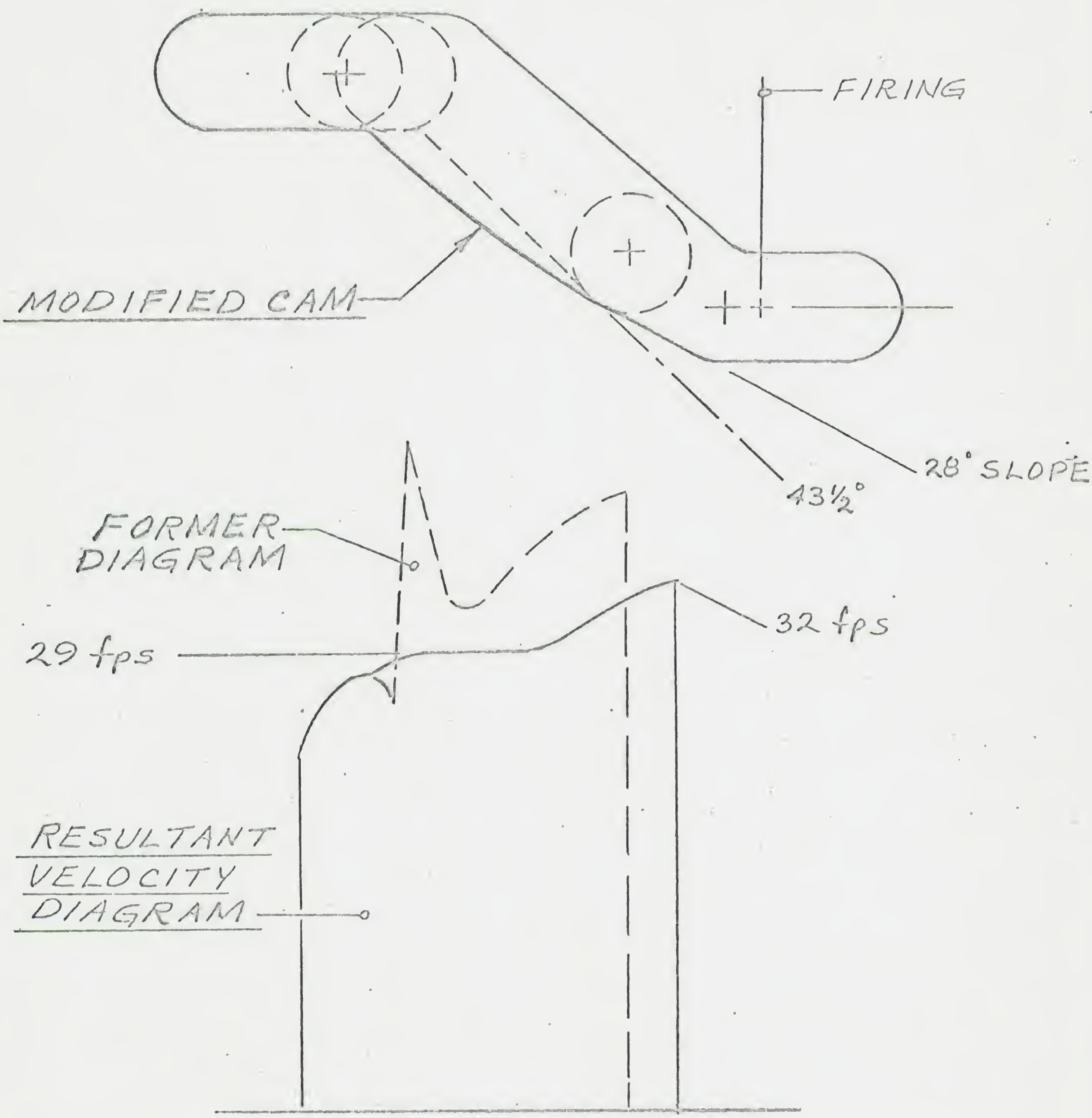
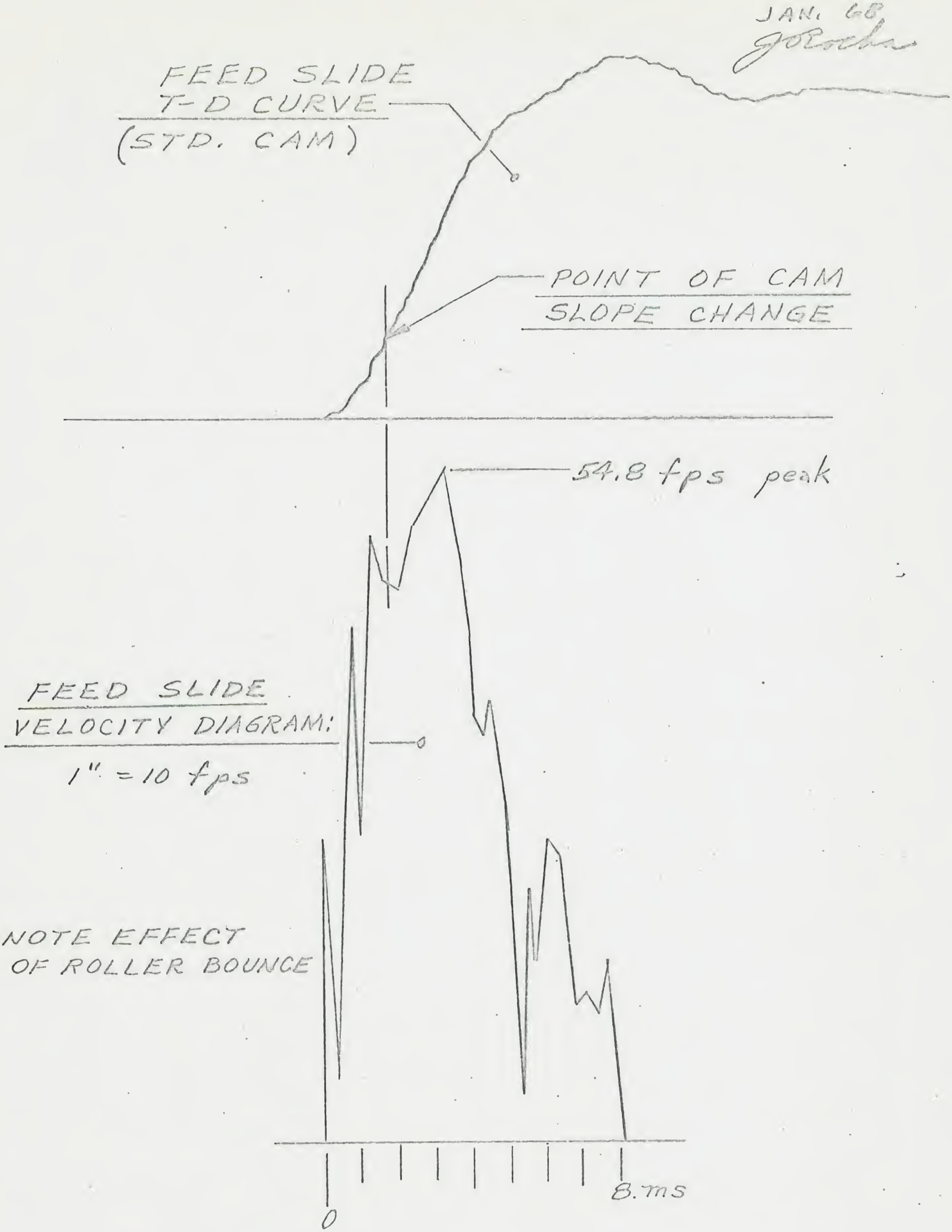


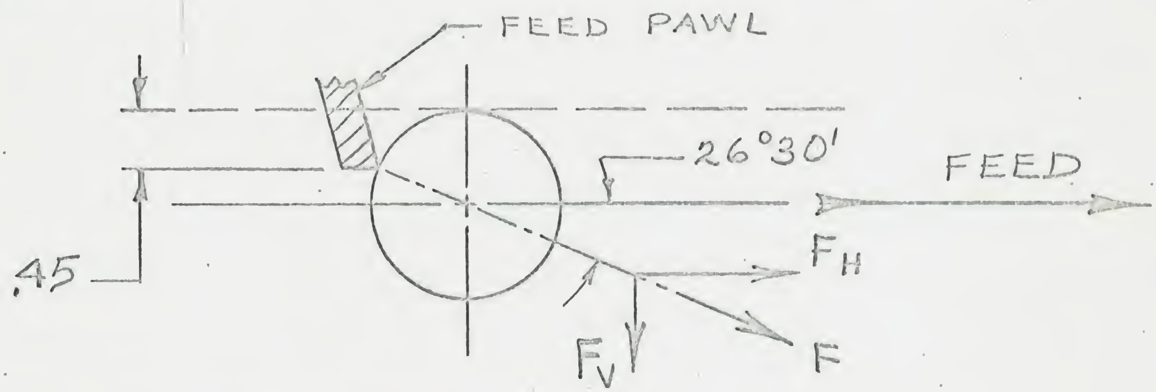
CAM MODIFICATION TO REDUCE INERTIA REACTION - FEED CAM "A"



FEED CAM LOAD DATA - "B"

The feed pawl should be pushing at a low point on the periphery. This will result in lower operating energies of the feed system, so that spring peak loads can be reduced, cocking loads modified to a more favorable level, and more normal feed forces realized.

The .45 inch engagement of feed pawl with cartridge should be increased to (.53 to .56) resulting in drive angles of 21° to 18°.



At the same time, the feed springs may be changed. This feed system is limited in belt pull by the load available at the end of the stroke. Therefore, the peak loads at the beginning of feeding may be reduced by using a spring of a lower rate.

In the present feeder, the springs have a load (at the end of the feed stroke) of 32 pounds, but in belt pull firing tests, can pull a load of only 13 pounds. This indicates an efficiency of only 40%, demanding the improvement in geometry noted above.

The following spring is recommended, compared with the present spring:

	Present	Recommended
Outside diameter	.575	.575
Wire diameter	.080	.070
Number of coils	43.	60.
Rate lb/in.	12.	4.5
Assembly load, lb.	16.	20.
Peak load, lb.	51.	33.

The bolt lugs should be re-designed to eliminate the high transverse load that is causing the bolt body to crack. The locking lug is an angled block that is positioned at an angle 18° to the horizontal. This causes a vertical component of force of 32% of the load to act in the direction that previous bolts have cracked.

The sharp inside rail corner should also be eliminated by a generous fillet. The sharp corner augments unnecessarily high stress concentrations.

The bolt cam plate should have both sides of the cam path parallel to each other, to prevent bolt lock bounce during function.

Further, the last round sensor should be removed. This is a device that holds the feed slide in a cocked position after the last round in the belt of ammunition is fired out. This device creates a safety hazard in that when one raises the cover, his fingers may well be in the slide path, and if the sensor is inadvertently touched, the slide is released with devastating results. This is very likely to occur during normal servicing. Functionally, the mechanism is not important.

Maximum recoil acceleration is shown as an 86° lift angle in the velocity diagram. This is equivalent to an acceleration of 42,000 fps^2.

Using F = m a, F = 13.03 X 42,000/g = 17,000 lb. This agrees with a chamber pressure of 10,000 psi, which develops a load of 20,000 lb.

Duration of the acceleration peak is .41 ms.

Also, considering that theoretically, recoil velocity is inversely proportional to bullet mass, we have:

$W_w V_w = W_p V_p$
V_p = 790 fps
W_p = .532 lb.
W_r = 13.03 lb.
V_r = .532 X 790/13.03 = 32.3 fps.

This should agree with the total turn-around velocity, which is 7.0 in counter-recoil and 25.48 in recoil, for a total of 32.48 fps.

Therefore observed results are in agreement with the general theory.

A further study of the critical feed slide / actuating mass reveals the following relationships:

a. With a feed slide of 1.1 lb., the reaction due to a 57,000 fps acceleration is 1.1 X 57,000/g X 1.95 = 3800 lb. (from item 8.)

b. The comparable barrel force at the same point:

$$W = 13.03 \text{ lb.}$$
$$V_1 = 24.6 \text{ fps}$$
$$V_2 = 23.0 \text{ fps}$$
$$t = .3 \text{ msec.}$$
$$\therefore \quad a = 5320 \text{ fps}^2$$
$$\therefore \quad F = m\,a = \underline{2160 \text{ lb.}}$$

This is why the barrel mass does not have sufficient force to drive the feed slide efficiently, but rather causes a sharp loss of energy in the primary mass.

c. The energy absorbed by the feeder during the period of high acceleration:

$$E = \frac{W}{2\,g}\ (24.6^2 - 23.0^2) = \underline{15.8 \text{ ft. - lb.}}$$

Barrel stroke = .0079 ft. during this period

$$F = 15.8/.0079 = \underline{2000 \text{ lb.}}$$

d. Energy transfer from barrel mass to feed slide (per T-D curve)

t = .00118 sec. (note increase from theoretical .0003 time)

$$E_s = \frac{W}{2\,g}\ d\,V = \frac{1.1}{64.4} \quad (53^2 - 45.3^2)$$

$$E_s = \underline{13. \text{ft. lb.}}$$

This agrees with the 15.8 ft.-lb. given up by the barrel.

e. An energy distribution chart was calculated for the eleven millisecond time period after firing. This is the critical portion of the cycle in which the area of inefficiency predominated:

Energy Distribution for 11 msec after firing:

t	Mass Energy	Diff	Slide Travel	Slide Energy	Spring Eg	Diff
1	45	+45.	-	-	-	-
2	123	+78.	0	2.45	-	+2.45
3	143	+20.	.35	26.	1.0	+24.5
4	98	-45.	.9	37.5	3.2	+13.7
5	62	-36.	1.55	46.0	6.3	+11.6
6	70	+8.	2.1	22.5	10.	-20.
7	81	+11.	2.44	9.8	12.5	-10.
8	88	+7.	2.62	3.3	13.8	-5.
9	94	+6.	2.8	.83	15.3	-1.
10	93	-1.	2.9	0	16.2	0
11	91	-2.	-			

Energy in ft. - lb.
Travel in inches

The most significant relationships are:

At ms (4), 31% of mass energy is transferred to feed system

At ms (5), 32% transferred to feed system

At ms (6), 40% of slide energy is transferred to mass (bolt & barrel)

Total energy released from mass to feed system = 49 ft. lb.

Total energy absorbed by feed springs = 16.2 ft. lb.

Efficiency = 33%

In conclusion; the weapon appears to be sensitive to the following combinations:

(a) Feed spring load vs. sear spring load,
(b) Drive spring load vs. sear plate spring load,
(c) Bolt bounce time at sear vs. time for feed pawls to engage round and start feed,
(d) Barrel unlocking time vs. time for cam plates to initiate bolt locking,
(e) Gouging between feed lever roller and bolt, causing sharp energy losses,
(f) Alignment of corner of link with cartridge stop rod.

These critical areas must be eliminated if the weapon is to be suitable for service.

18. Summarizing the recommendations offered:

(a.) Improve feed cocking cam contour
(b.) Feed pawls should be lowered
(c.) Feed springs should be changed to reduce peak load.
(d.) Use stranded wire for drive springs
(e.) Radius the corners of the feed arm roller
(f.) Improve bolt lock geometry
(g.) Hard-coat guideways in aluminum receiver. (In the long run, a steel receiver would be more favorable)
(h.) Remove last-round sensor
(i.) Relocate anti-surge pawl to a position nearer the bolt.
(j.) Finally, avoid sharp, re-entrant corners through the weapon design.

The Soviet AK-47 Assault Rifle

Brief History of Soviet Automatic Rifle Development

Prior to the Soviet activity in automatic rifle systems, the standard bolt action rifle was the 7.62mm 1891 Mosin-Nagant, in several rifle and carbine versions. Advances in automatic weapon development led to the 6.5mm 1916 Federov, the 7.62mm Simonov Automatic Rifle of 1936 and the 7.62mm Tokarev semi-automatic rifle of 1940. The Tokarev is quite similar in principle to the Belgian FN rifle.

These weapons did not prove to be suitable for service, so the bulk of infantry weapony in World War II featured the PPD and PPSh series of sub-machine guns chambered for the Type P-7.62mm Pistol Cartridge with a muzzle velocity of 1640 fps. Longer range firing required extensive use of heavy machine guns.

Accordingly, since the effective range of the sub-machine guns was limited, a family of weapons built around a more effective, but lightweight, cartridge was required. This led to the development of the 7.62mm M43 cartridge that apparently was a counter-part of the 7.92 German Kurz (short) cartridge. The effective range is in the order of 440 yards. Evidently, the trade-off between "firepower" as a matter of number of rounds carried and "firepower" as a matter of effective range, was well considered.

The submachine guns of World War II are no longer used in the Soviet armies, being replaced by the AK-47 types. A number of satellite armies still retain the sub-machine guns, of economic necessity.

As discussed previously, the following analysis is only visual, as instrumented firing records were not available.

Analysis of Soviet AK-47 Assault Rifle

The Soviet 7.62mm Model AK-47 is a magazine fed, percussion fired, gas operated, semi/full automatic weapon. In its geometry, it has a slight drop-stock, pistol grip, with the gas piston located above the barrel, ramp-type open sights, with a battle range setting of 300 meters, and a mid-height front sight.

The thirty-two round magazine curves forward, to suit the accumulated taper of 32 rounds, and extends approximately 7.0 inches below the receiver.

The selector lever also functions as a receiver dust cover when the selector is in the safe position. This provides good visual reference for the shooter as well as "touch" reference at night-time.

In general appearance, with a short barrel and pistol grip, it may be first considered as a sub-machine gun, but the 7.62mm cartridge places it nearer the rifle class.

Its principal characteristics, compared with the U.S. M14 rifle are as follows:

	AK 47	M 14
Overall length	34.5 in. (no flash supp.)	44 in. (with flash supp.)
Barrel length	16.37 in.	22.0 in.
Weight without magazine	8.4 lb.	8.2 lb.
Magazine weight	1.0 lb.	.5 lb.
Magazine capacity	32 rounds	20 rounds
Fire type	semi/full auto	semi auto (selector optional)
Bolt carrier dwell	.35 in.	.40 in.
Travel to end of unlock	.68 in.	.94 in.
Total bolt carrier travel	5.32 in.	
Bolt rotation to unlock	35°	

Ammunition Characterisitics:

	M43 (AK 47)	NATO (M 14)
Caliber	7.62mm	7.62mm
Round weight	253 grains	375 grains
Round length	2.18 in.	2.8 in.
Case length	1.52 in.	2.01 in.
Projectile weight	122 grains	150 grains

	M43 (AK 47)	NATO (M14)
Charge weight	25 grains	47 grains
Muzzle velocity	2329 fps	2800 fps
Effective range	440 yards	660 yards

There are five variations of the Soviet M43 cartridge (1) Ball type with mild steel core, (2) Tracer type T-45, (3) Armor piercing, incendiary, (4) Incendiary tracer, and (5) Blank, with rosette style crimp.

Additional AK-47 Technical Data:

Weight of reciprocating masses:

Bolt assembly	.198 lb.
Bolt carrier assembly	.99 lb.
Drive spring assembly (.066/2)	.033 lb.
Total	1.221 lb.
Ratio of primary mass to secondary mass	5.01/1

Firing Mechanism:

Firing pin weight	.0139 lb.
Hammer weight	.101 lb.
Mass Ratio	7.27/1.0

Gas System

Piston diameter	.548 in.
Piston travel before bleed	.70 in.

Gas regulator: none required

Rate of Fire Approx. 600 shots/min.

The breech mechanism consists of a rotating bolt actuated by a reciprocating bolt carrier. The bolt carrier rides in keyways in the receiver. An internal cam is machined in the forward section of the bolt carrier and rotates the bolt during the locking and unlocking phases of the cycle. Two locking lugs are positioned at the front of the bolt and are diametrically opposed. The lock cam lug is mounted on the outside periphery of one of the bolt lugs, increasing the moment arm for a favorable cam force leverage. This mechanism is a further development of the U.S. M1 rifle bolt mechanism. Note the highly favorable mass ratio between bolt carrier and bolt. (5/1)

A slender cylindrical section of the bolt body is supported in the bolt carrier. The bolt also contains a free-floating firing pin and an extremely simple cylindrical extractor.

A single drive spring, mounted in a telescoping guide rod, drives the bolt carrier assembly in counter-recoil. The guide rod base also functions as a cover latch; therefore the spring serves double duty. Also, when the spring assembly unit is removed, it remains as an easily handled sub-assembly. The receiver housing may be removed and the weapon function, for visual inspection of the operating mechanism, may be studied.

The gas system is of the plain impingement system, with the piston being an integral part of the bolt carrier. The piston end is concave, as is the end of the gas piston housing. This provides an initial chamber volume. The upper handguard is also the piston housing, with gas bleed holes incorporated in the gas cylinder extension. A single gas orifice is used, with no adjustment for power necessary. The gas piston is ribbed, for rigidity, and the operating rod is easily accessible for the bolt cam cuts.

The receiver also functions as the firing mechanism housing assembly. The firing mechanism is not a modular unit, as in the M1 rifle. The firing mechanism has eleven parts, including three retaining pins. The automatic sear spring has a single long arm that groove-locks these pins. Three sears are used in this mechanism with a double claw hammer for the primary and secondary sears, and a single (hammer hub) notch for the automatic sear, actuated by the operating rod. The primary and secondary sears are identical to the M1 in principle. When the hammer is in battery position, the safety can be applied. This would cause a jam when attempting to charge the weapon. However, the charging slot closure is a good visual indicator that the safety is on.

The selector shaft controls the functioning of the semi-automatic sear and trigger.

The front sight is a hooded post which can be adjusted by using the combination tool in the tool kit, either by screwing up or down, or moving left or right. The rear sight is of the conventional V-notch tangent leaf, the sight radius being approximately 15 inches. The upper forearm is retained by a latch on the rear sight. When the upper forearm is removed, a latch for the lower forearm is revealed. The lower forearm conceals a hiding pocket in the receiver.

The curved magazine tube is made of heavy gauge spot-welded construction with critical areas, such as the feed lips and catches, being machined. The magazine follower is a stamping, with a long skirt to control tipping, by its close fit with the inside wall of the magazine.

The magazine contains a number of highly desirable design features. The extremely rugged magazine lips are most favorable for extended field use. The magazine follower does not have to be critically balanced between ammunition stack and follower spring. No matter where one bears down on the follower (the center, forward, or rearward positions) the follower moves in the magazine tube smoothly. The spring design, therefore can be simple oval coils, free of stress concentrations, and free from binding along the magazine ribs.

The magazine follower design, together with the 5/1 mass ratio between bolt carrier/bolt are the two reasons why this weapon continues to fire in the field with old, corroded, apparently unusable, ammunition.

A hole at the lower rear surface of the magazine tube is an excellent visual indicator that the magazine is full. The user merely adds rounds to the magazine until the bottom round shows up in the hole.

A three piece tool kit is mounted in the buttstock, with a spring-biased pressure plate facilitating entry and removal. The kit contains a combination tool, a bore brush, and a cleaning patch prod, or jag. The combination tool provides a screw driver blade, a punch, and two wrenches. The cleaning rod is stored under the barrel and through the lower hand guard. The body of the tool kit is a tool handle, and the cap can be attached to the muzzle of the barrel as a guide for the cleaning rod, and, presumably, as a blank firing attachment.

The cycle of operation is quite identical, in principle, to the M 14 rifle, except that a fixed ejector, integral to the receiver, is used.

Weapon field stripping is accomplished without tools, by a system of guide slots in the receiver, for the operating rod, and retaining latches for the upper and lower hand guards.

The barrel is rifled with 4 lands and grooves, with a right hand twist. The muzzle attachment nut is threaded left-hand. The gas port is at an angle, which simplifies cleaning. No parts other than the gas piston and upper forearm are removed for this bit of maintenance. Training in weapon maintenance is considered to be fairly simple. While not convenient for bayonet fighting, the short weapon length is handy for street and house-to-house fighting. The basic weapon is also equipped with a folding buttstock, for paratroop and special services.

Note that no flash suppressor is used, in spite of the short (16.3 in.) barrel; however the powder charge is low, being only 25 grains.

An estimate of production cost was made, and based on a quantity of a lot of 2 million rifles, the weapon would cost approximately $60.00 without product engineering or final inspection services. Also, it was estimated that approximately 550 machine operations are necessary, as against 800 for the M14 rifle. The weapon is almost completely made from milled steel components, with few stampings being made. This is a reversal of form, since most World War II Soviet weapons used stampings en masse. The receiver has relatively few complex milling cuts, and an insert is used to cam the bolt into the barrel extension at the start of the locking rotation.

This weapon is standard issue not only in the Soviet Union, but also in the satellite countries of East Germany, Rumania, Hungary (with modified handguard and plastic pistol grips) as well as in Communist China, which designates their production as 7.62mm type 56 assault rifle. This is supplied to North Vietnam also. The Czech assault rifle is similar in outward appearance, but is redesigned internally to a different mechanism.

In summary, the AK-47 weapon employs a compact, essentially well-designed bolt mechanism, with the action suited to the short 7.62mm cartridge. Generally, this cartridge is considered in Europe as the "mid-'30" cartridge, that is, of energy levels mid-way between the U.S. Cal. .30 carbine and cal. .30'06 cartridges. It is considered as a further development of the German 7.92 mm Kurz (short) cartridge. However, for machine guns, the Soviets still retain their old 7.62mm rimmed full length cartridge (as well as in accuracy match rifles).

AK-47 accuracy is nearer rifle class than sub-machine gun class, as should be expected. The average submachine gun firing single shots will produce a group of from 12 to 18 inches diameter at 100 yards. The AK 47 will group in 6 inches at 100 yards. In full automatic fire, the weapon climbs rapidly, when firing bursts of 5 or more rounds, therefore a good gripping position on the forestock and sling is necessary.

The weapon fires from the closed bolt position for either the semi automatic or full automatic cycles of operation. There is no bolt-hold-open device to hold the action open after the last round in the magazine is fired.

The AK-47 will eventually be replaced by the AKM, a modification which is characterized principally by a sheet metal receiver, rather than the milled receiver, as well as several other minor changes.

VII Bibliography and Recommended Reading

Much of the material discussed in this manuscript is taken from data developed at Springfield Armory. All of the material is unclassified.

Many of the chapters are augmented by supporting data taken from several Ordnance libraries. A bibliography lists the books used, together with an identifying subscript which correlates the sources with the corresponding chapters of the manuscript.

For additional data, on any of the topics listed, the volumes noted should be used for reference, among others.

The designation "SA" indicates topics in which Springfield Armory data was prominent. Certain chapters, notably on "Firing Mechanism Design" and "Feeding" contain only SA data. Much of this has not been documented previously, and is knowledge acquired through "on-the-job-training" at Springfield Armory.

BIBLIOGRAPHY

Reference		Author
a	"Small Arms of the World"	Smith & Smith
b	"Hatcher's Notebook"	J. S. Hatcher
c	"The Machine Gun" (4 Vol.)	G. M. Chinn
d	"Treasury of the Gun"	H. L. Peterson
e.	"Oerlikon Pocket Book"	Oerlikon Works
f	"Principles of Firearms"	C. E. Balleisen
g	"Elements of Ordnance"	Hayes
h	Winchester Ammunition Handbook	O M C C
j	"Ballistics"	Cummings
k	"Cartridges of the World"	F.C. Barnes
l	"Ordnance"	published bi-monthly
m	"The Bullet's Flight from Powder to Target"	F. W. Mann
n	"The Gun and its Development"	W. W. Greener
p	"World War I"	S. L. A. Marshall
q	"Handbook of Mech Spring Design"	Associated Spring Corp
r	"Spring Design & Application"	N. Chironis
s	"Cams"	H. A. Rothbart
t	"Formulas for Stress & Strain"	R. J. Roark
u	Speer Reloading Manual #77	Speer, Inc.
v	NRA Fact Book (Firearms & Ammo)	NRA
w	"Gun Digest"	published annually
x	"Shooter's Bible"	published annually

Bibliographical Reference	TOPIC IN ORDER OF CONTENT
	1. Introduction
	2. Historical
d,e,n,v	Development of the Military Cartridge Case
SA,a	U.S. Light Rifle Program
SA, 1	Caseless and Liquid Propellant Systems
	3. Supporting Sciences
SA,b,c,e,f,k,m,u,v	Interior Ballistics
SA,b,e,h,u	Exterior Ballistics
SA,b,e,g,v,w	Recoil
SA,1,(Jan 1964)	Dynamics of Automatic Rifles
	4. Systems of Operation
SA,f,c,	Blowback
SA,e	Recoil Operation
SA,e	Gas Operation and Analysis
	5. Weapon Design
SA,b,v,	Headspace
SA	Factors of Safety
SA,t	Barrel and Bolt Lug Stresses
SA	Firing Mechanism Design
SA	Feeding
SA,c	Link Design
SA	Magazine Design
SA,q,r	Spring Desing
j,p,b,f,v	Muzzle Devices
	6. Weapon Analysis
SA,c	Time - Displacement Curves
SA,s,t	Analysis of recoil-operated machine gun
a,1	Analysis of Soviet AK-47
	7. Bibliography & Recommended Reading
	8. Courses of Study

SA designates "Springfield Armory"

VIII Courses of Study

In becoming knowledgeable as to Small Arms Ordnance principles and practice, there are a number of sources of information available to the design engineer. These include the following:

a. Hardware and Literature Survey

b. On-the-job training

c. Extension courses

The "hardware and literature survey" includes both museum and library facilities and will familiarize the engineer with many of the facets of ordnance engineering conducted in the past. Much engineering has been done in the past that was not successfully concluded because of production limitations, metallurgical limitations, and other requirements that would not pose a barrier today. The physical principles employed are the important elements, rather than the outward appearance.

On-the-job-training is highly specialized and time-consuming training process. The more knowlegeable an engineer is about prior art, the less likely he is vulnerable to making a false start. However, he should not be limited to prior art, as quite often a bold new approach will result in marked improvements in performance. Proper supervision in "on-the-job-training" stabilizes the engineer and the program.

Extension courses are most valuable in augmenting both prior art studies and on-the-job-training. These are available to engineering personnel and usually are free of charge. The Ordnance School conducts a wide variety of these courses; the ones of principle interest are as follows:

	Credit Hours
ORD 4 - Fundamentals of Ballistics	16
ORD 411 (61) Machine Guns	19
ORD 413 (63) Hand & Shoulder Weapons	14
ORD 508 (111) Research & Development	10
ORD 601 Weapons Familiarization	5
ORD 605 (60) Principles of Small Arms	13
ORD 606 (69) Armament Principles	12
ORD 713 Aircraft Armament Subsystems	2

Personnel may apply for these, and other, courses by filing a DA Form #145, and in block #7 address:

Commanding Officer
Rock Island Arsenal
Rock Island, Illinois 61201
ATTN: SWERI-PTT-2430

or through the training branch of any other appropriate installation.

Other courses of interest include those in an engineering curriculum such as: Strength of Materials, Kinematics, Dynamics, Metallurgy, etc. with "Strength of Materials" considered as the most important one by this writer.